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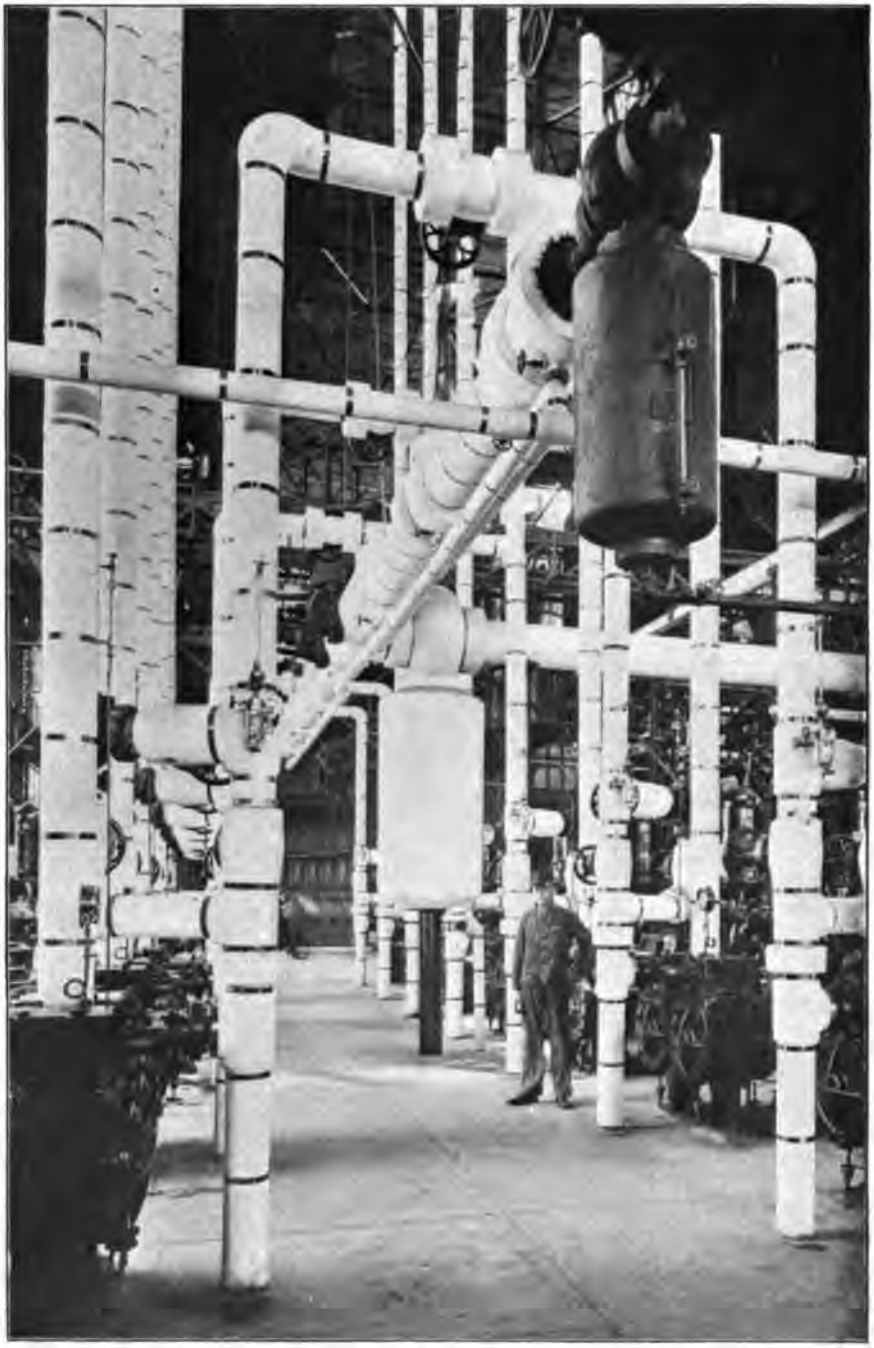

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The Philip Carey Mfg. Co

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A General Reference Work

ON MACHINE SHOP PRACTICE, TOOL MAKING, FORGING, PATTERN MAKING,
FOUNDRY WORK, METALLURGY. STEAM BOILERS AND ENGINES,
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Grateful acknowledgment is here made also for the invaluable co-operation of the foremost manufacturers and engineering firms, in making these volumes thoroughly representative of the best and latest practice in the design and construction of steam and gas engines, machine tools, and other classes of modern machinery; also for the valuable drawings and data, suggestions, criticisms, and other courtesies.

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Foreword

THE rapid advances made in recent years in the field of Mechanical Engineering, as seen in the evolution of improved types of machinery, new mechanical processes and methods, and even new materials of workmanship, have created a distinct necessity for an authoritative work of general reference embodying the accumulated results of modern experience and the latest approved practice. The Cyclopedia of Mechanical Engineering is designed to fill this acknowledged need.

¶ The aim of the publishers has been to create a work which, while adequate to meet all demands of the technically trained expert, will appeal equally to the self-taught practical man, who, as a result of the unavoidable conditions of his environment, may be denied the advantages of training at a resident technical school. The Cyclopedia not only covers the fundamentals that underlie all mechanical engineering, but places the reader in direct contact with the experience of teachers fresh from practical work, thus putting him abreast of the latest progress and furnishing him that adjustment to advanced modern needs and conditions which is a necessity even to the technical graduate.

¶ The Cyclopedia of Mechanical Engineering is based upon the method which the American School of Correspondence has developed and successfully used for many years in teaching the principles and practice of engineering in its different branches. It is a compilation of representative Instruction Books of the

School, and forms a simple, practical, concise, and convenient reference work for the shop, the library, the school, and the home.

¶ The success which the American School of Correspondence has attained as a factor in the machinery of modern technical and scientific education, is in itself the best possible guarantee for the present work. Therefore, while these volumes are a marked innovation in technical literature—representing, as they do, the best ideas and methods of a large number of *different* authors, each an acknowledged authority in his work—they are by no means an experiment, but are in fact based on what has proved itself to be the most successful method yet devised for the education of the busy workingman. They have been prepared only after the most careful study of modern needs as developed under the conditions of actual practice in the Machine Shop, the Engine Room, the Drafting Room, the Factory, etc. The formulæ of the higher mathematics have been avoided as far as possible, and every care exercised to elucidate the text by abundant and appropriate illustrations.

¶ Numerous examples for practice are inserted at intervals; these, with the test questions, help the reader to fix in mind the essential points, thus combining the advantages of a textbook with those of a reference work.

¶ Grateful acknowledgment is due the corps of editors and authors—engineers and designers of wide practical experience, and teachers of well-recognized ability—without whose co-operation this work would have been impossible.

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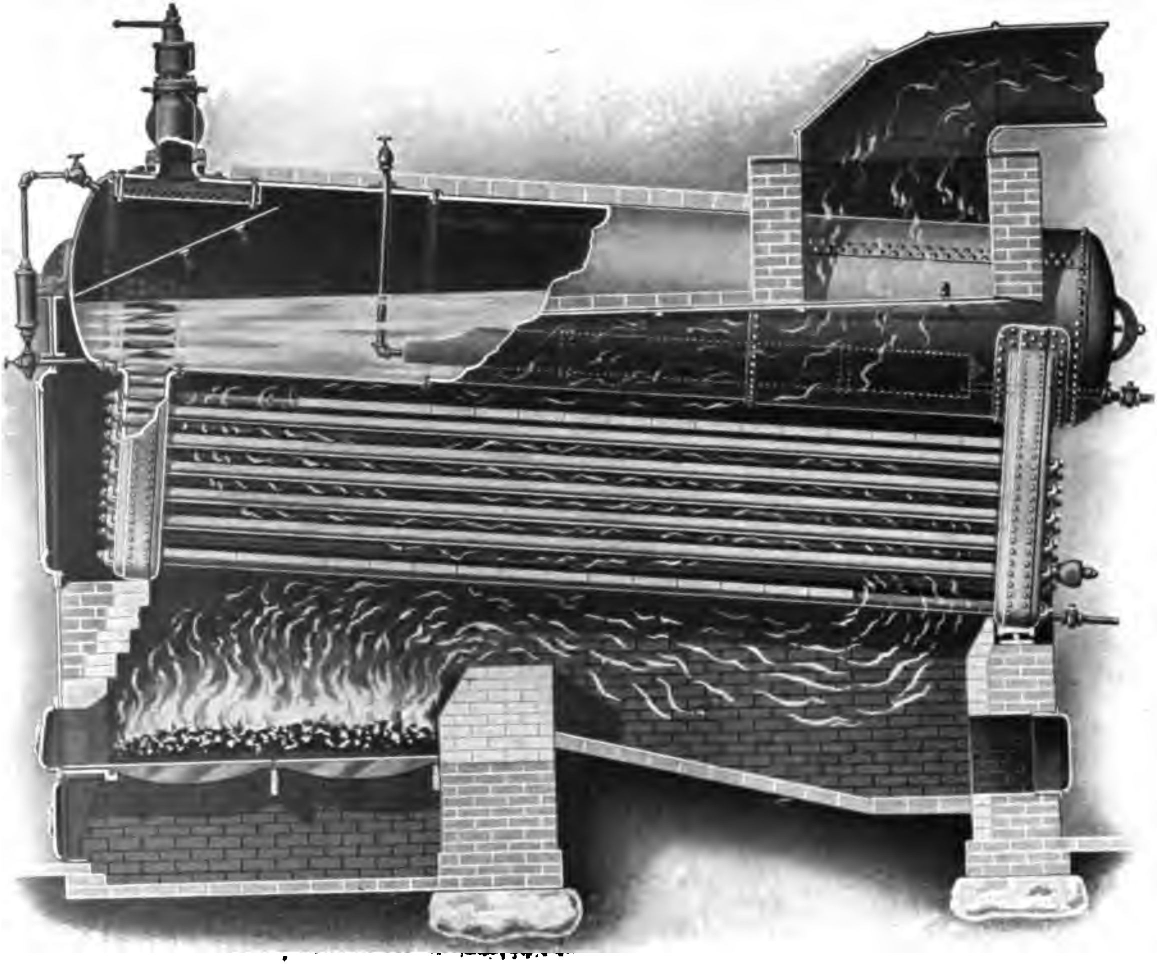


FRANKLIN WATER TUBE BOILER.

CONSTRUCTION OF BOILERS.

A steam boiler, or steam generator, consists of a vessel to contain the water and the steam after it is formed; a fire-box to contain the fire; tubes, flues and uptake to transmit heat and conduct the hot gases from the fire to the chimney, and various fittings to facilitate the safe and economical operation. Boilers are often classified according to their uses and conditions; thus we have stationary, marine and locomotive boilers. Boilers having a shell partially filled with tubes, through which the hot gases pass, are called tubular, fire-tube or shell boilers; and those having a large flue in which is placed the fire, are called flue boilers. If the tubes are filled with water and the hot gases are outside, the boiler is called a water-tube boiler.

Steam boilers are made in a variety of shapes, according to the type, uses and conditions. Let us first consider boiler construction in general, leaving out the peculiarities of marine, locomotive and water-tube boilers.

MATERIALS.

The materials of which boilers are constructed are exposed to conditions which weaken them and shorten the life of the boiler. Among these conditions are corrosion, both external and internal, high pressure, and expansion and contraction, due to varying temperature and pressure.

Cast iron was the material of which the earliest forms of boilers were made, but on account of its low tensile strength and its unreliable nature, it is now but little used, except for parts of water-tube boilers, and sometimes for the ends of low-pressure cylindrical boilers and for fittings. It is cheap and resists corrosion but on account of its unreliability and brittleness, the parts must be made thick and therefore heavy.

Wrought iron, up to about 1870, was the principal material used for boiler plates. It is a pure iron prepared from pig iron by a process called puddling, described in "Metallurgy." Wrought iron is well adapted for use in boiler construction, as it is strong, tough and fibrous, and combines high tensile strength with ductility and freedom from brittleness. When the properties mentioned are well combined, wrought iron will resist strains due to unequal expansion. Boiler fastenings, stays and other parts made by welding are sometimes made of wrought iron. It is customary to consider that a bar loses about one-quarter of its strength by welding, although it is often stronger in the weld, owing to the working of the metal during the welding process.

Steel has entirely displaced iron for boiler-shell work. Boiler steel is made by the open-hearth process, and contains for ordinary thickness of 1 or $1\frac{1}{4}$ inches 0.25 per cent carbon, while thinner plates of $\frac{1}{4}$ inch should not contain over 0.15 per cent carbon. Larger percentages of carbon, while accompanied by an increase in tensile strength, lessen the ductility. The following properties show steel to be the best boiler material at present: great tensile strength, ductility, homogeneity, toughness, freedom from blisters and internal unsoundness. Blisters and unsoundness are faults sometimes met with in wrought-iron plates.

Copper in many respects is superior to wrought iron for boiler construction. It is homogeneous, resists oxidation (the corrosive action of most feed waters) and incrustation. It is more ductile and malleable and a better conductor of heat, which not only gives it a higher evaporative power, but also enables it to last longer under the intense heat of the furnace. Its disadvantages are its low tensile strength, about 30,000 pounds per square inch, and its decrease of strength with an increase of temperature. In heating from the freezing point to the boiling point it loses 5 per cent of its strength, and at 550° F. it loses about one-quarter of its strength. For these reasons and on account of its high price, it is now seldom used in boiler work.

Brass is an alloy of copper and zinc in which the proportions of each vary considerably. The red color comes from a larger per cent of copper. Red brass is better and more expensive than yellow brass. Brass is used for valves, gauges and other fittings.

Bronze is an alloy of copper and tin, and is advantageously used for valves and seats of safety valves where the wear is great.

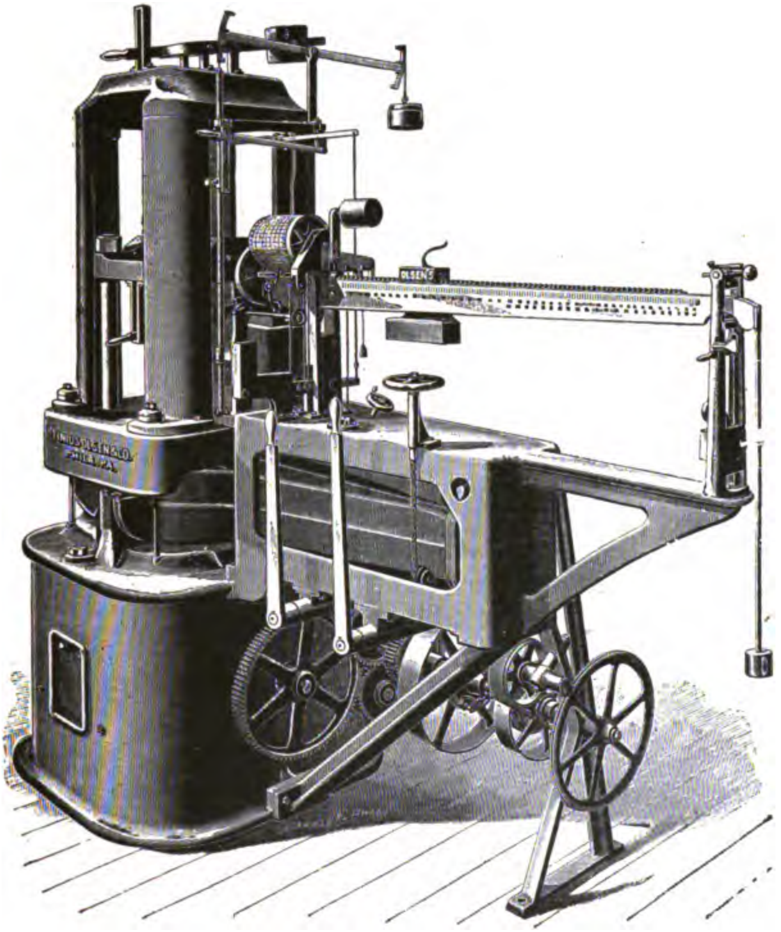


Fig. 1.

TESTING MATERIALS.

In order to determine the strength and the other qualities of the materials, specimens are tested. The results of these tests show the ultimate tensile strength, elastic limit, contraction of area and elongation.

The simplest way to test a piece of iron bar or plate would be to fix it firmly at the upper end and hang weights on the other end, adding other weights until the bar is broken. This is but a crude method, and in order that the elastic limit and elongation may be determined at the same time, testing machines are used. There is a large variety of testing machines, adapted for various materials, but the general principles are the same.

Testing Machines. The testing machine consists of a frame and two heads, to which the ends of the test piece are fastened by wedges or other devices. By means of steam or hydraulic power one head is drawn away from the other for tensile tests. The pull is transmitted to some weighing device, usually levers and knife edges like the beam of ordinary platform scales. In small machines the pull may be applied by a lever.

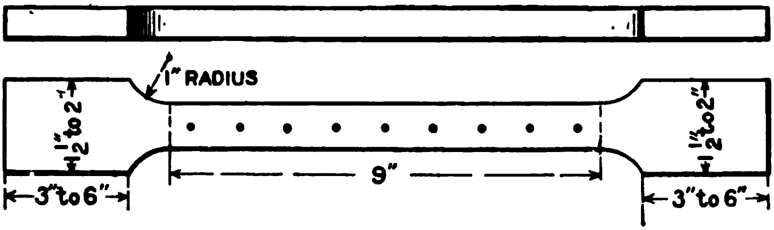


Fig. 2.

Testing machines are made for all varieties of testing: tensile, compressive and shearing stresses. Also for deflection of beams and for strength of wood, cement, brick and stone. Fig. 1 shows an Olsen testing machine designed for tensile and compressive tests of iron and steel.

In order to test materials, test pieces or specimens are prepared. For testing iron plate the test piece should be at least 1 inch wide, about 2 feet long and planed on both edges. Many engineers recommend these dimensions. According to the Board of Supervising Inspectors of Steam Vessels, the test piece should be 10 inches long, 2 inches wide and cut out at the center.

To ascertain the tensile strength and other qualities of steel, a test piece should be taken from each plate. These test pieces are made in the form as shown in Fig. 2. The straight part in the center is 9 inches long and 1 inch wide; and to determine elongation it is marked with light prickpunch marks at distances 1 inch apart, the marked space being 8 inches in length. The ends are $1\frac{1}{2}$ inches to 2 inches broad and 3 inches to 6 inches long.

As has been explained in "Mechanics," the force necessary to break the piece is the proportionate part of the tensile strength per square inch. Thus if the test piece having a reduced section of .4 square inch is broken at 19,200 pounds, the tensile strength of the plate is $\frac{19{,}200}{.4} = 48{,}000$ pounds per square inch.

EXAMPLES FOR PRACTICE.

1. If a piece of boiler plate breaks at 33,500 pounds and the reduced section is $1\frac{1}{8}$ inches by $\frac{1}{2}$ inch, what is the ultimate tensile strength?

Ans. 59,555 pounds.

2. A boiler plate is claimed to be of 64,000 pounds tensile strength. If the section is 1 inch wide and .63 inch thick, what should be the reading of the testing machine when the specimen breaks?

Ans. 40,320 pounds.

3. A test piece of the form shown in Fig. 2 measured 8 inches between the prickpunch marks before testing and 9.56 inches after testing. What was the per cent of elongation?

Ans. $19\frac{1}{2}$ per cent.

4. If the area of section before breaking is .4825 square inch and after breaking is .236 square inch, what is the per cent of reduced area?

Ans. 51 per cent.

STRENGTH OF BOILER MATERIALS.

The crushing strength of cast iron is high, varying from 50,000 to 75,000 pounds per square inch; its tensile strength is low, varying with the chemical and physical properties of the iron from about 15,000 to 22,000 pounds per square inch.

Wrought-iron plates having a tensile strength of from 50,000 to 60,000 pounds, with an elongation or ductility of from 20 per cent to 30 per cent, are suitable for boiler work. Boiler iron may be tested in the following ways if testing machines are not available: Cut from the plate a strip about 2 inches wide and bend it cold, down upon itself; if it shows no fracture on the outside curve, it is satisfactory. This is, however, a severe test, and only the best flange iron will stand it; on the other hand, any iron which, when heated to a cherry red and bent, shows cracks or fracture on the outer curve, is unfit for use in boiler construction. When wrought iron was used for boiler plates it was customary to give the plate what is called the hammer test. The plate was suspended clear of the ground and struck with a hammer at intervals of three or four inches over its surface; a clear, ringing tone indicating a sound plate, while a dull sound indicated with fair certainty a defect such as internal unsoundness.

Mild steel has a tensile strength of from 55,000 to 65,000 pounds per square inch, with an elongation of 25 per cent. A test piece cut from a plate $\frac{3}{4}$ inch thick or less should stand bending double, when hot or cold, and not show any cracks; thicker plates should be capable of being bent at a small radius to a large angle without showing any cracks. Steel should never be worked at a blue heat, as in this state it is very brittle. It is also mechanically tested by being heated to a cherry red, quenched in water at 82° F. then bent in a curve of small radius; if it cracks, it has become tempered, and it is therefore unsuitable for this work. If the tensile strength of the steel is under 70,000 pounds per square inch, it is sufficiently tough and ductile and can be easily worked.

In general, boiler materials are carefully tested for the following qualities:

Tensile strength, to resist rupturing strains. Also in order that the plates may be thin.

Toughness and elasticity, to resist corrosion and the wear and tear of manufacture.

Ductility, so that the boiler may change its shape slightly without rupture. This is a more important quality.

BOILER CONSTRUCTION IN DETAIL.

The drawing or design of the boiler is worked out in the draughting room, as explained later under the head of Boiler Design.

The draught shows the general arrangement of the boiler, together with complete detail drawings, from which the materials are ordered. These materials are plates, rods for stays, rivets, stay bolts, tubes, steel bars, angles and channel bars for stiffening, etc.

In some boiler shops it is customary to lay the boiler out on a large blackboard full size, thereby checking the drawing. In ordering plates the blank forms are filled out in the following manner:

Messrs. John Blank & Co:

Please furnish us with the following Steel Plates, Ultimate Tensile Strength, *60,000;* Elongation, *25 per cent:*

Number wanted.	Thickness	Dimensions.	Marks.	Remarks.
6	1″	90″×70″	S 14	**Shell**

The dimension which runs in the direction the plate is to be bent is given first. The plates are marked as per order blank, and this serves to identify the plate when the occasion arises. When ordering any odd shape, a sketch with dimensions must be placed in the column headed "Remarks."

In ordering plates, allow for trimming, particularly in the case of irregular shapes. Rivets are sold by the pound, regardless of their shape or size. Round and flat iron may be ordered by the running foot. Manufacturers publish tables showing weight of rivets, round iron, etc., with which they furnish boiler makers.

Boiler shops are equipped with the following tools: plate rolls, plate planers, shears, drill presses, punches, countersinking machines, flanging machines, hydraulic and steam riveters, and a compressed-air system for operating pneumatic machines, such as calkers and chippers. They also have machine shops for doing such machine work as is required for fittings, furnace fronts, etc., and a system of cranes for handling and transporting material. In connection with the above is a storeroom of sufficient size, a forge shop, and an engine and boiler for supplying the shop with the power necessary to operate it.

In boiler-shell work drilling has entirely displaced punching, and to-day all holes are drilled. Punching is cheaper than drilling, but it is more injurious to the plates and not as accurate. It is easy to see that drilling rivet holes, even if twenty are being drilled at once, is done with less strain on the plates than when done by a multiple punch forcing several holes at once. The force required to punch a plate gives the best idea of the harm done to the plate. Experiment shows that the resistance of a plate to punching is about the same as its resistance to tensile tearing. Suppose this to be 50,000 pounds per square inch; then the force required to punch the plate is the area cut out times the shearing strength, or $d \times \pi \times t \times 50{,}000$.

In which formula

d = diameter in inches and

t = thickness in inches.

For a hole $\frac{3}{4}$ inch in diameter in a $\frac{1}{2}$-inch plate, the force will be

$$\tfrac{3}{4} \times 3.1416 \times \tfrac{1}{2} \times 50{,}000 = 58{,}900 \text{ pounds.}$$

If the force required to punch one hole is 58,900 pounds, the force required in punching several holes by means of a multiple punch is enormous.

A good, ductile plate is but little injured by punching; but if of a hard, steely nature, it is likely to be seriously injured. For this reason wrought-iron plates are usually punched and steel plates are drilled. On the whole, a drilled plate is somewhat stronger than a punched plate for any kind of joint.

Some boiler makers punch the rivet holes slightly smaller than the desired size and then ream them out. By this process the injured metal around the holes is cut away. Another method to overcome the injurious effects is to anneal the plate after punching.

The ordinary process of annealing consists of heating the plate to red heat, and then allowing it to cool slowly. By this means, hard and brittle iron or steel is made soft and tough. While the metal is hot, the surface becomes oxidized. For most purposes this scale of oxide in not harmful, but in some cases it must be removed. As this is expensive, a process of annealing in illuminating gas has been devised. The action of the gas is to reduce the oxide without altering the properties of the piece. The results obtained from annealing depend upon the kind of iron or steel, the temperature to which it is raised, and the rate of cooling. It is a great advantage to all steel of over 64,000 pounds per square inch in tensile strength, but softer steels are little better for the process.

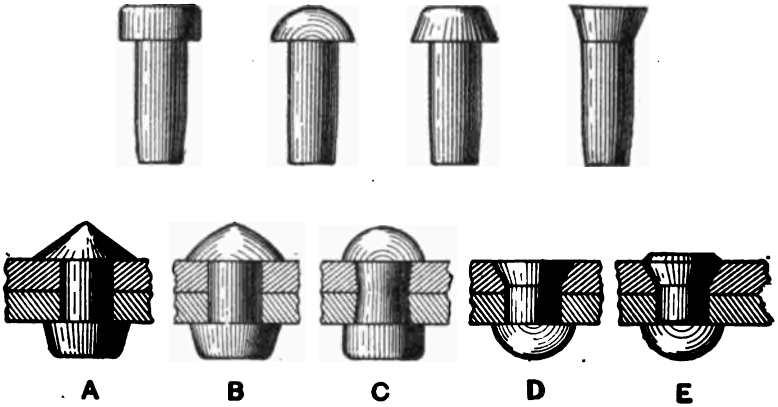


Fig. 3.

After the shell plates are planed to correct shape and the holes drilled or punched, they are put through the bending rolls and bent into a cylindrical shape, the amount of curvature being determined by a template made for the purpose. Plates are usually sheared to size, and then the edges planed with a slight bevel to facilitate calking. In the meantime the heads are being flanged by a hydraulic flanging machine; when the flange is completed, the head is put on the platen of a boring mill and turned so as to exactly fit into the shell. In some shops it is customary to punch or drill only a few holes in the shell and flange of the head, these holes serving to take bolts for holding the parts together. The back head plate is bolted into the rear course of plating, and the parts thus assembled are hoisted up to drill if the plates, etc., have not been previously drilled or punched, otherwise to the hydraulic riveter.

RIVETS AND RIVETING.

Rivets are formed by forging, from round iron bar or mild steel, with a cup or pan shaped head. The cylindrical part, called the shank, is a little smaller than the hole and has a slight taper. Fig. 3 shows common forms of rivets. As rivets are not as reliable in tension as in shear, they are used mainly at right angles to the straining force. If the stress is parallel to the axis, bolts are used, since they are strong in tension. The shearing strength of steel rivets is about 45,000 pounds per square inch, and of iron rivets about 40,000 pounds per square inch. Steel rivets are often used with steel plates, but many boiler makers prefer to use iron rivets in all cases.

Three types of rivets in use are shown in Fig. 4, the following table giving the dimensions:

Diameter of Rivet.	Cone Head. A			Countersunk. B		Button Head. C	
D	E	F	G	E	G	E	G
$\frac{5}{8}$	$1\frac{1}{16}$	$\frac{19}{32}$	$\frac{9}{16}$	$1\frac{1}{16}$	$\frac{9}{32}$	$1\frac{1}{16}$	$\frac{7}{16}$
$\frac{11}{16}$	$1\frac{1}{8}$	$\frac{21}{32}$	$\frac{39}{64}$	$1\frac{3}{16}$	$\frac{5}{16}$	$1\frac{1}{8}$	$\frac{1}{2}$
$\frac{3}{4}$	$1\frac{1}{4}$	$\frac{23}{32}$	$\frac{21}{32}$	$1\frac{1}{4}$	$\frac{3}{8}$	$1\frac{1}{4}$	$\frac{9}{16}$
$\frac{7}{8}$	$1\frac{7}{16}$	$\frac{13}{16}$	$\frac{3}{4}$	$1\frac{3}{8}$	$\frac{7}{16}$	$1\frac{7}{16}$	$\frac{3}{4}$
1	$1\frac{5}{8}$	$\frac{15}{16}$	$\frac{27}{32}$	$1\frac{5}{8}$	$\frac{1}{2}$	$1\frac{5}{8}$	$\frac{3}{4}$

Formerly all joints of boilers were riveted by hand, but now all riveting is done by machines, except those joints to which a machine cannot be applied. If done by hand, the red-hot rivet is inserted in the hole, and the second head formed by two riveters working with hammers. This head is either made conical by the hammers alone or finished with a cup-shaped die called a "snap." This latter is the more usual method. The disadvantages of hand riveting are slowness and a tendency to form a shoulder before the rivet fills the hole.

Machine riveting is preferable, as the work is done better, faster and more accurately; the pressure coming gradually on the entire rivet, compresses it completely into the hole before the head is formed. Before riveting, care should be taken that the plates are close together, so that a shoulder will not be formed between the plates and prevent a good joint. Rivets should always be put in while red hot, for in this condition they are more easily worked, and when they cool they contract, nipping the plates together in a tight joint.

Hydraulic riveting is more gradual and is generally preferred to steam riveting. The pressure from the steam riveter often comes as a sudden blow and does not allow time for the rivet to completely fill the hole.

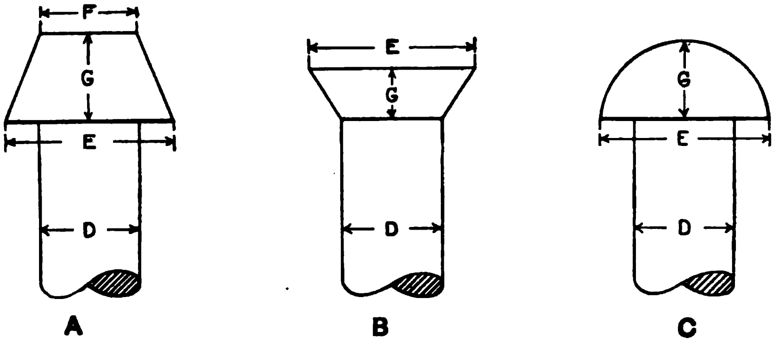


Fig. 4.

It is sometimes desirable to rivet with a countersunk head; that is, the rivet does not project above the plate. The countersunk head is formed by hammering down the end of the rivet into the countersink in the plate. This form is shown at D, Fig. 3. This joint is often used in shipbuilding and in boiler making when it is necessary to attach mountings. It should always be avoided, if possible, on account of its weakness, and especially when the straining force acts in the direction of the length of the rivet, as the head has a very insecure hold and is likely to be pulled through the hole.

Rivets may be tested in a boiler shop as follows: the rivet to be bent cold in the form of a hook around another rivet of the same diameter, and show no flaws or cracks; to be bent hot down upon itself and show no cracks, head to be flattened while hot until its diameter is 2½ times the diameter of the shank, and show no flaws.

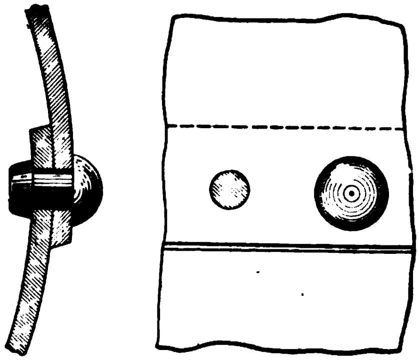


Fig. 5.

The uniform heating of steel rivets is of more importance than in the case of iron rivets, where it is sufficient to heat the points only. Steel rivets also should not be heated to a white heat, as iron rivets are, but to a bright cherry red, for if heated beyond this point they will burn. The fire in which steel rivets are heated should be kept thick, and the draught moderate. This should also be observed in heating steel plates for flanging.

There are various forms and strengths of riveted joints. It is obvious that in punching or drilling, a plate is weakened to the extent of the sectional area cut out, and that if the holes are punched, the metal between the holes is weakened. In treating the strength of a joint it is customary to speak of it as a percentage of the strength of an unpunched plate.

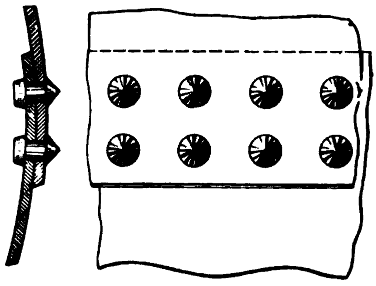


Fig. 6.

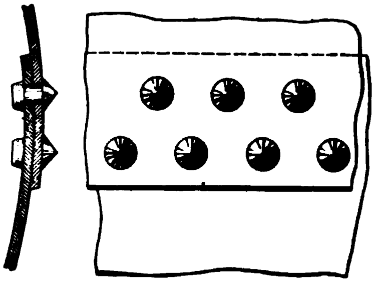


Fig. 7.

If one plate overlaps another and is riveted to it by a single row of rivets, as shown in Fig. 5, it is called a single-riveted lap joint. This joint has about 56 per cent of the strength of a solid plate. If another row of rivets is added, it is called a double-riveted lap joint; Fig. 6 shows the double-riveted lap joint chain riveted, and Fig. 7 the double-riveted lap joint zigzag riveted.

Double riveting is done in two ways: zigzag, or staggered, and chain. When rivets are put in so that the rivets of one row are opposite the spaces of another row, it is called zigzag riveting or staggered riveting. If the rivets are placed immediately opposite each other, it is called chain riveting.

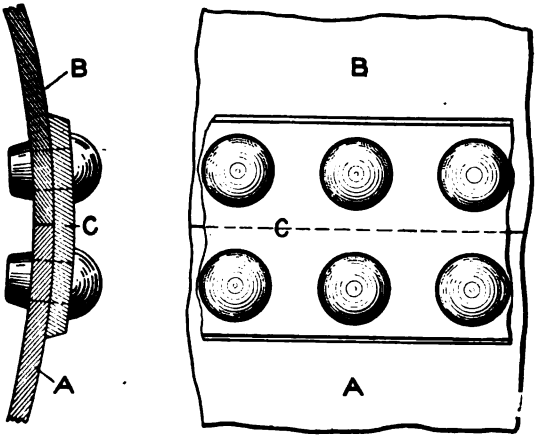


Fig. 8.

If the two plates are kept in the same plane and a cover or butt strap riveted on, it is called butt riveting (Fig. 8, in which A and B are the boiler plates, and C is the butt strap). If an inside butt strap is added, it is called a double butt joint (Fig. 9). Fig. 10 shows a treble-riveted butt joint. A single butt joint is about equal in strength to a lap joint having but one row of rivets, but a double butt joint is considerably stronger.

In this latter form of joint the rivets have double shearing surfaces, since they tend to shear off in two planes. This either makes a stronger joint or allows the use of smaller rivets. In the single butt joint the butt strap is usually about $1\frac{1}{8}$ the thickness of the plate, and if the inside butt strap is added, each butt strap is made about ⅝ the plate thickness. Butt joints are now being used in the best class of boilers, and are used almost entirely for plates less than ½ inch in thickness.

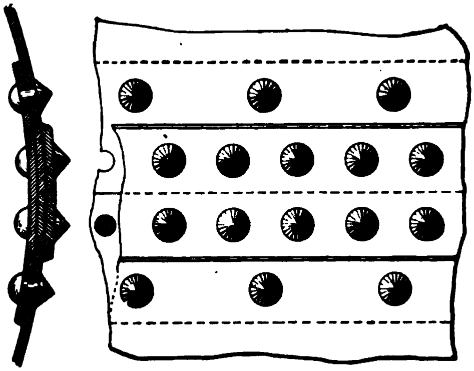


Fig. 9.

Lap joints are used for circumferential seams, and the stronger joint, the butt, for longitudinal joints. For high pressures in marine boilers, triple riveting is frequently used.

If a cover plate is riveted on the outside of a lap joint, it is called combined lap and butt joint. In this case there are three rows of rivets, the middle row having twice as many rivets as the outer rows. Fig. 11 shows the combined joint.

The distance between the centers of rivets is called the "pitch." The mathematical calculation of pitch and the distance between the rivets and the edge of the plate will be taken up later.

The following table gives an idea of the relative strengths of riveted joints:

Kind of Joint.	Riveting.	Percentage of Strength.	
		Punch.	Drilled.
Lap	Single Double	55 69	62 75
Single Butt	Single Double	55 69	62 75
Double Butt	Single Double	57 72	67 79

FLANGING IRON AND STEEL PLATES.

Iron plates are more severely tested by flanging than by any other work done upon them. This is due to their fibrous nature, and great care is necessary to prevent breaking in the bend, if the corner is sharp.

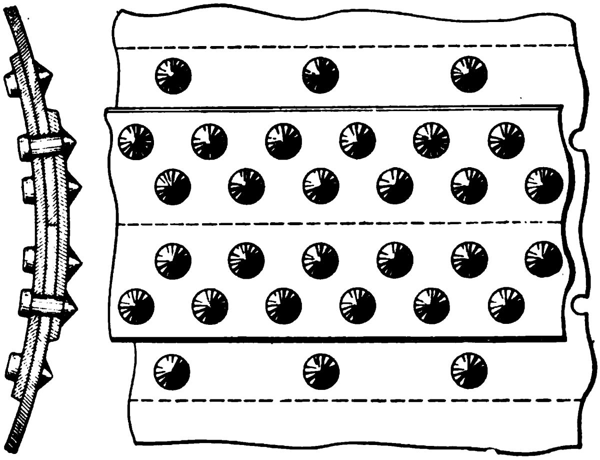


Fig. 10.

As has been stated, steel requires uniform heating and moderate curves. Flanging is almost entirely done to-day by machines. After flanging, the steel should be annealed by heating the whole plate uniformly to a dull red heat, and allowing it to cool slowly.

WELDED JOINTS.

Welded joints for boiler shells are desirable. By their use deposits which accumulate on and around rivet heads and joints, corrosion caused by leakage, and loose rivets, are done away with, and calking also. Moreover, a perfectly welded joint is stronger than the best riveted joint, and approximates nearly to the original strength of the plate. Welded steam drums are used now quite extensively for water-tube boilers of the marine type.

The soundness of such a joint is a matter of uncertainty, and depends upon the skill and care of the workmen. It is impossible, from external appearances, to judge the soundness of a welded joint. The principal use of welded joints is for furnace tubes and steam domes, but they have not been used much for boiler shells. The lack of tests on welded joints and the small amount of information on the subject, render the results of experiments of little value. The weld is best made when the edges of the plates are upset, at red heat, to nearly double the plate thickness, and beveled to an angle of about 45 degrees. The edges are then heated together, and the weld made by hammering down the joint to the original thickness of the plate.

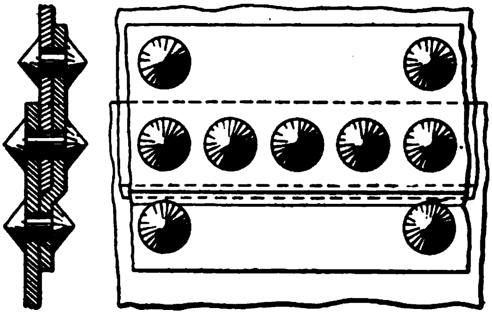


Fig. 11.

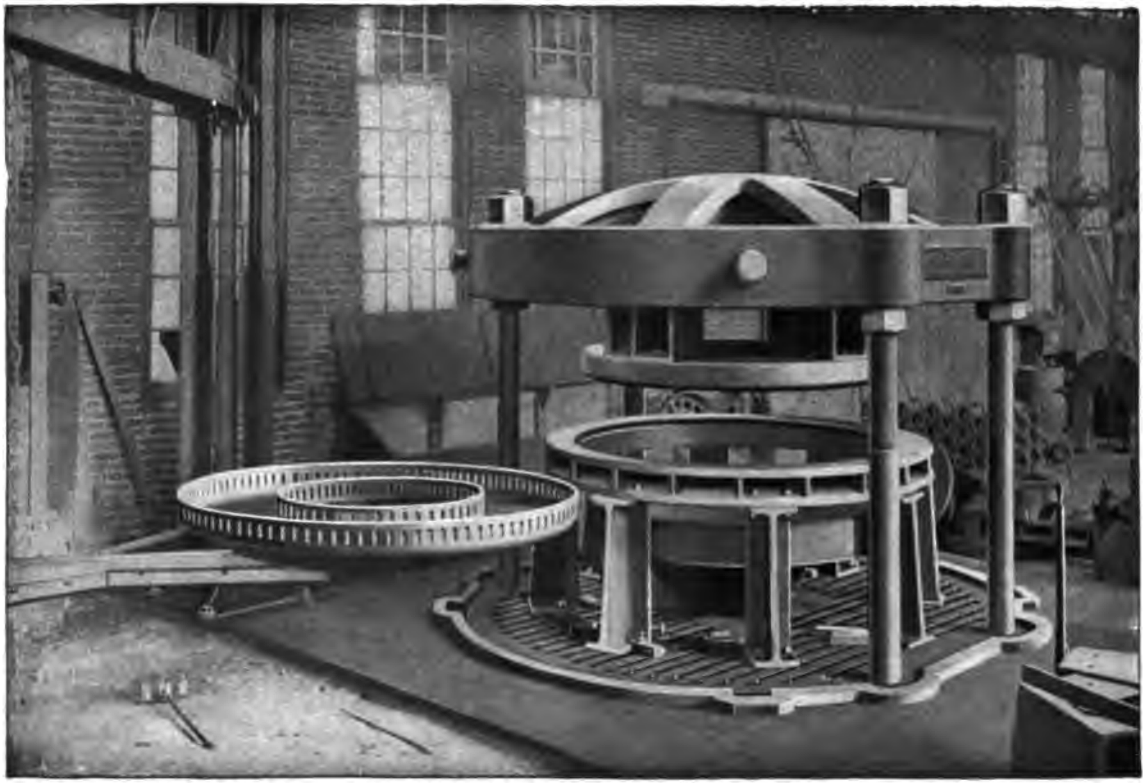
ARRANGEMENTS OF PLATES AND JOINTS.

When we take up the design of boilers we shall see that a boiler tends to rupture longitudinally. The reason for this is that the resistance of a thin cylinder to circumferential rupture is double the resistance to longitudinal. Since this is the case, lap joints are used for transverse seams, and a stronger form (the double butt joint) is used for the longitudinal.

At the junction of three or more plates, where the circumferential and longitudinal joints meet, ordinary riveted joints would be too thick. To overcome this difficulty, two or more plates are forged thin at the joint, as shown in Fig. 12.

Whenever longitudinal and girth seams meet, the plates should be arranged to "break joints"; that is, one longitudinal seam should not be a continuation of another. The proper arrangement is shown in Fig. 13.

In both vertical and horizontal boilers the inside lap is made to face downward, so that it will not form a ledge for the collection of sediment



HYDRAULIC FLANGING PRESS. "CAHALL" FACTORY.

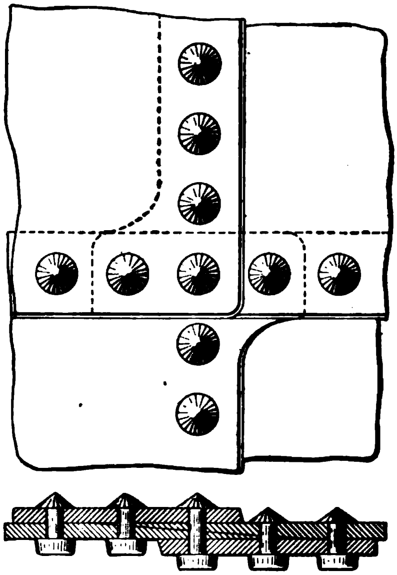


Fig. 12.

The belts of plates that make up the length are sometimes arranged conically, with the outside lap facing backward. When the boiler is slightly inclined toward the front end, this conical arrangement facilitates draining and cleaning, as the dirt is removed at the front end. This is a great advantage to internally fired boilers, as they are difficult to clean.

In long vertical boilers the ring seams are arranged with the inside lap facing downward, so as not to have a ledge for sediment. Sometimes the belts of locomotive boilers are arranged telescopically, with the largest diameter at the fire-box end. Of late years the best makers use larger plates than formerly. This is advantageous, especially in externally fired multitubular boilers, as the single seam is placed above the water-level, and therefore is away from the fire.

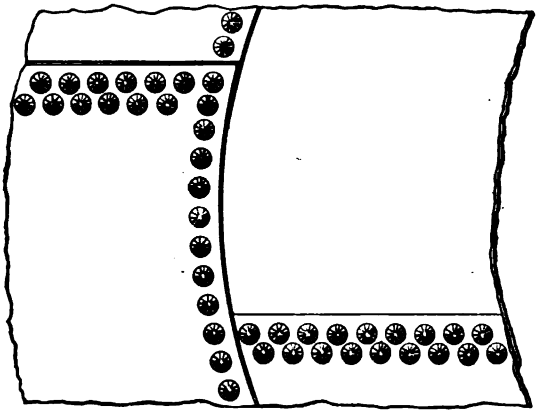


Fig. 13.

The portion of a boiler between the shell and the furnace is called the water leg. Figs. 14 to 20 inclusive illustrate the method of construction of the water leg and the joints around the furnace door. Figs. 14 and 15 show two methods of constructing

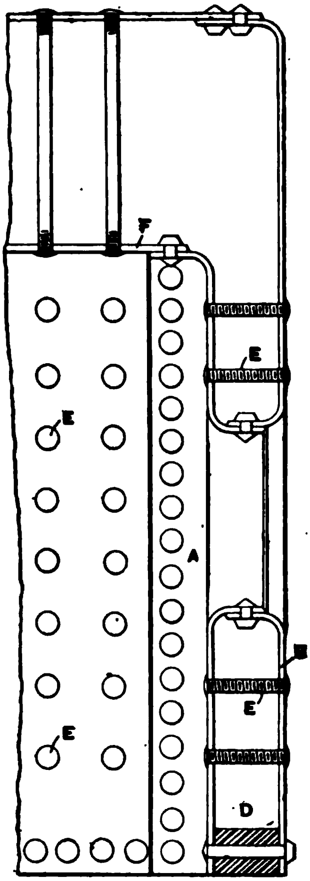


Fig. 14.

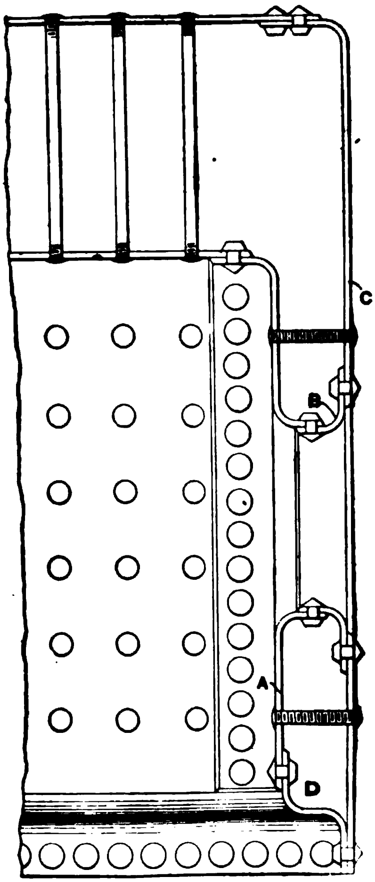


Fig. 15.

the water leg. In Fig. 14 the exterior plate and the furnace plate are riveted to the ring D by means of long rivets. This ring is usually made of wrought iron, but in many cheap boilers it is of cast iron. In Fig. 15 the two plates are riveted to the flanged ring D. This construction is better than the solid cast-iron ring, on account of flexibility, but the junction of the plates D and C forms a corner in which sediment is deposited. In Fig. 17 the plate B is flanged and riveted to C. This arrangement requires less riveting than the one shown in Fig. 15. Figs. 14, 15 and 17 also show three forms of construction of the joints around the furnace door. In Fig. 14 both the exterior plate and the furnace sheet are flanged and riveted together. This is shown in an enlarged view in Fig. 18. The construction shown in Figs. 15 and 19 is not as good as that in Fig. 14, because of the extra riveting; also, it has two corners, B and C, for the deposit of sediment. Fig. 17 shows a somewhat different form of furnace construction,

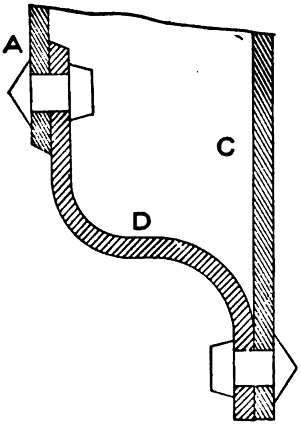


Fig. 16.

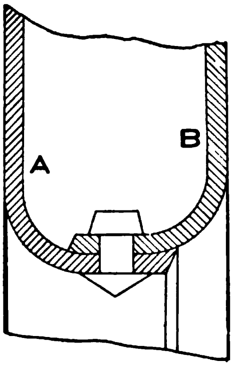


Fig. 18.

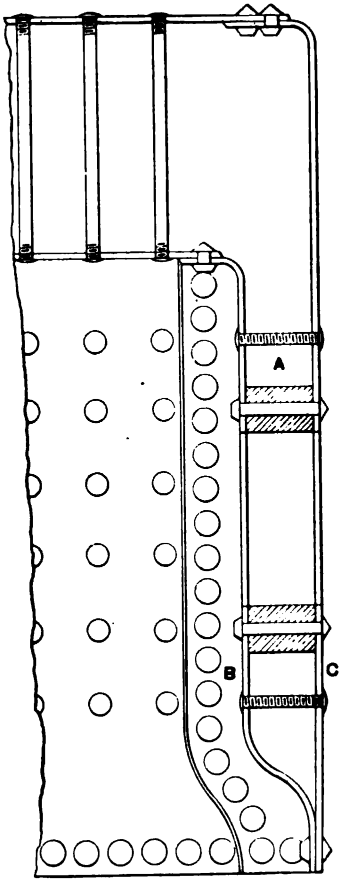


Fig. 17.

the two plates being riveted to the cast-iron ring. This form is better shown in Fig. 20. It makes this part of the boiler too rigid, but it has the advantage of not having rivet heads to wear off. In these methods of riveting, those which have the flanged ring are preferable to those using the cast-iron ring, because of more freedom for expansion; but the flanged ring forms an undesirable corner.

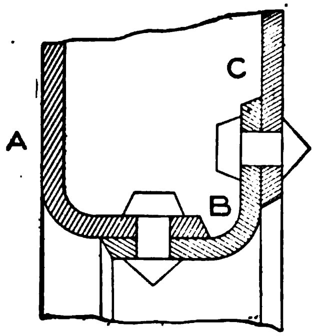


Fig. 19.

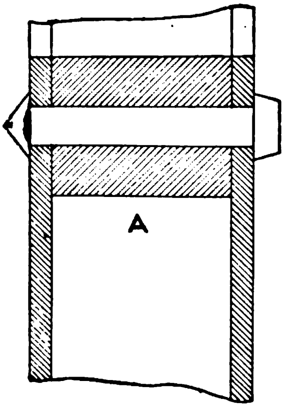


Fig. 20.

In almost every boiler, plates must be connected at right angles. An example of this is seen where the end plates are jointed to the shell plates of cylindrical boilers. There are three principal methods: riveting both plates to an angle iron, riveting to a flanged ring and flanging the end plate. In Fig. 21 the two plates are riveted to an angle iron, which is made of wrought or cast iron. This construction is too rigid; the constant variations of temperature cause repeated changes of form, which tend to crack the angle iron on the inside of the plate at the joint. Corrosion increases the evil, as it rapidly attacks iron which has once been cracked or broken. There is no definite rule for the dimensions of these angle irons, but it is safe to make the mean thickness a little greater than that of the plates.

The forms shown in Figs. 22 and 23 are better. The head is flanged and riveted to the shell plates. The flanging makes a more flexible joint. The radius of the curve of the flange should be about four times the thickness of the plate. The head and shell are sometimes connected to a flanged ring, as shown in Fig. 24. The extra row of rivets makes a complex joint.

In vertical boilers the external fire-box is joined to the cylindrical shell by riveted joints. Figs. 25 and 26 show two forms; that in Fig. 25 being the better on account of the flanged ring, which allows expansion and contraction of the shell and furnace plates.

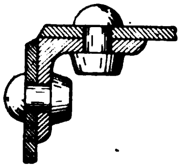


Fig. 21.

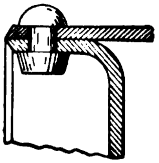


Fig. 22.

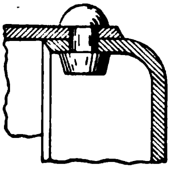


Fig. 23.

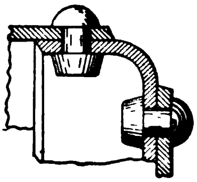


Fig. 24.

Sometimes the case occurs of connecting two plates which are parallel and near together. For instance, at the bottom of the locomotive fire-box a connection must be made between the inner and outer fire-box. The water-leg construction is a similar case. Several methods for this construction are shown in Fig. 27. Fig. 27A is too complicated and is undesirable, both on account of the numerous rivets and angle irons, and on account of the inside joints, which cannot be calked. Fig. 27B is better, since it has but one angle iron; it has, however, the undesirable inside joint.

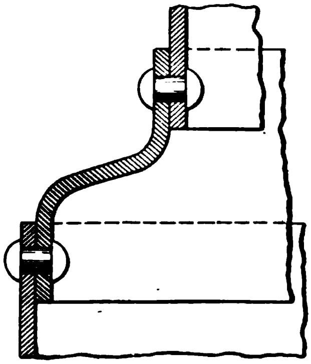


Fig. 25.

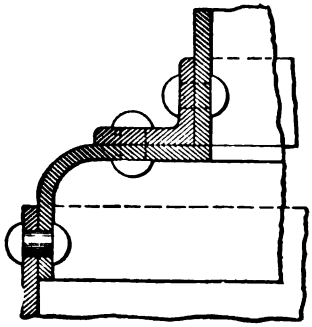


Fig. 26.

Fig. 27D is a good joint, the form of connection being called a channel iron. Fig. 27E, as we have seen, is a good flexible joint, but it has the undesirable corner where sediment lodges.

We have thus briefly discussed the various methods and arrangements for putting shells together, and now let us return to our boiler, which is ready for riveting at the hydraulic riveter. A few rivets are first driven at equal intervals around the ring seam at the back head. The reason for driving only a few rivets is that any errors in the spacing of the holes are distributed and not accumulated, as would be the case if they were driven in succession. From this point on, the riveting is continued until the shell is completely riveted up.

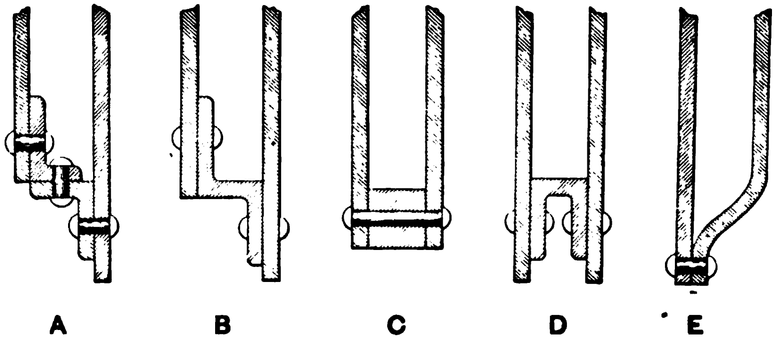


Fig. 27.

STAYING.

The shell is now ready to receive the stays. When under steam, a cylindrical shell is strained by internal pressure in two directions, namely: transversely, by a circumferential strain due to the pressure tending to burst the shell by enlarging its circumference, and longitudinally, by the pressure on the ends. If a boiler were spherical it would require no stays, because a sphere subjected to internal pressure tends to enlarge but not to change its shape. All flat surfaces in boilers must be stayed, otherwise the internal pressure would bulge them out and tend to make them spherical in shape. The ends of steam drums on high-pressure water-tube boilers are often made hemispherical.

The first and most important point in staying is to have a sufficient number of stays so that they will entirely support the plate without regard to its own stiffness. The second is to have them so placed as to present the least obstruction to a free inspection, and third, to have them so arranged as to allow a free circulation of water. Too much care cannot be taken in fitting stays and braces, as they are out of sight for long periods, and a knowledge of their exact condition is not always easily obtained. In the ordinary fire-tube boiler the principal surfaces stayed are: the flat ends, crown sheets, flat sides of locomotive boilers and combustion chambers of cylindrical marine boilers. In the case of most marine or Scotch boilers, the diameter is large compared to the length; hence the flat surface is considerable, and needs careful staying. All the plates that are not cylindrical or hemispherical must be stayed. The details should be arranged for each boiler; a few general methods and cautions can, however, be given.

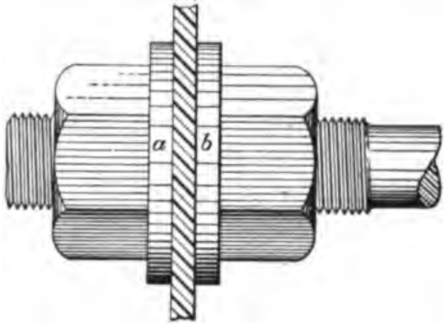


Fig. 28.

The most common and simple form of stay is a plain rod. It is used to stay the flat ends of short boilers. This stay is a plain rod passing through the steam space and having the ends fastened to the heads. The ends are fastened and the length adjusted in a variety of methods; the simplest being nuts on both sides of the plate, as shown in Fig. 28. The copper washers *a* and *b* strengthen the plate and prevent abrasion by the nuts. In place of the nuts the rod is often bolted to angle irons, which are riveted to the plates. In this case, turn buckles similar to the one shown in Fig. 29 are used for adjusting the length.

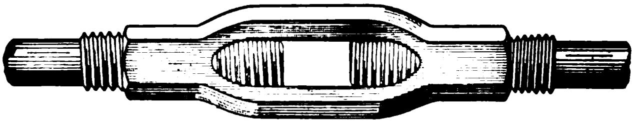


Fig. 29.

The stays are usually from $\frac{3}{4}$ inch to an inch in diameter, and are made of wrought iron or steel, with an allowable stress of 5,000 to 7,000 pounds per square inch. If the ends are fastened to riveted angle irons, the combined area of the rivets is made a little greater than that of the rod.

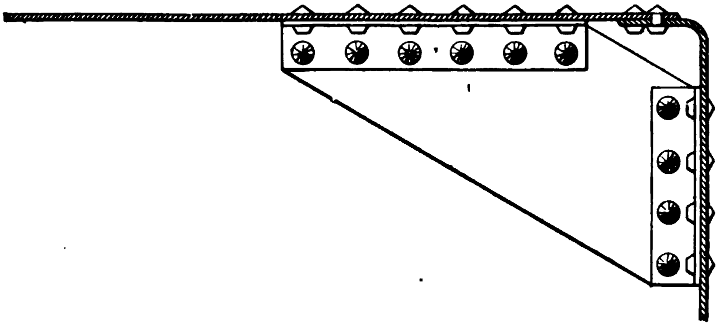


Fig. 30.

If a boiler is long, that is, more than 20 feet, long stays would sag in the middle and not take up the full stress on the end plates. For long boilers, gusset and diagonal stays are used. This form of boiler stay, shown in Fig. 30, is made of wrought-iron plate riveted to angle irons; the angle irons being riveted to the end and shell. Boilers of the Cornish, Lancashire and Galloway types often have this kind of stay. These boilers are internally fired, and as the variation of temperature causes expansion and contraction, great care should be used in placing the gusset

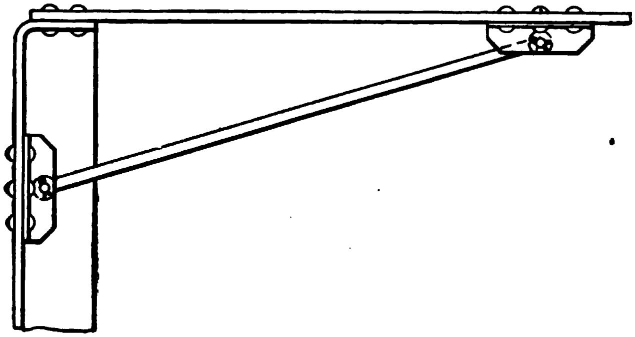


Fig. 31.

stay. If the stay is too near the flange or too many stays are used, the head will be too rigid and have a tendency to crack.

A form of diagonal stay is shown in Fig. 31. The plain rod is connected to angle irons by means of split pins. The angle irons are fastened to the shell and end by rivets or bolts. Another form of diagonal stay, called the crowfoot, is shown in Fig. 32. The two ends are bolted or riveted to the end and shell.

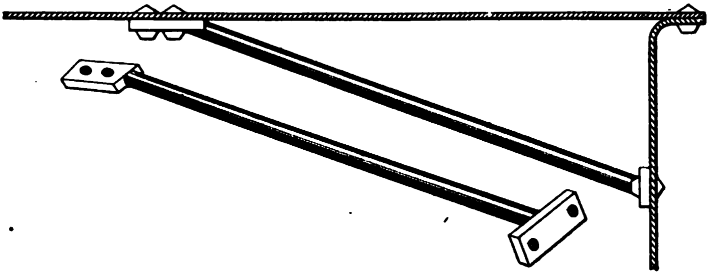


Fig. 32.

The angle between the shell plate and stay rod should be small,— not more than 30 degrees. The rod itself is designed for tensile strength, since the diagonal pull may be easily reduced to an equivalent direct pull. A large factor of safety is used to provide for future corrosion.

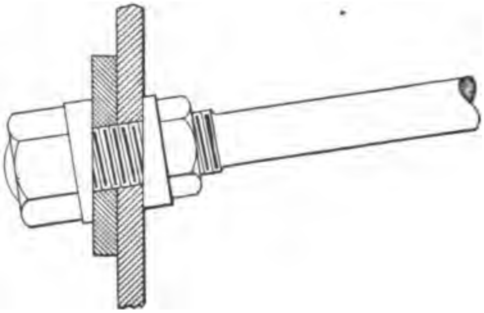


Fig. 33.

For marine boilers, a modified crowfoot stay (Fig. 33) is often used. The end passing through the head is supplied with nuts and taper washers, the washers having the proper taper to allow the nuts to be set up tightly against them.

In locomotive fire-boxes and in the combustion chamber of marine boilers, there are two flat or slightly curved surfaces that must be stayed together. These are riveted by short screw stay bolts. The bolts shown in Figs. 34 and 35 are screwed in place, and the ends riveted over. In marine boilers these stays are fastened with nuts, as shown in Fig. 36, instead of being riveted.

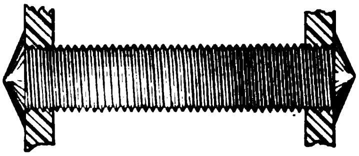


Fig. 34.

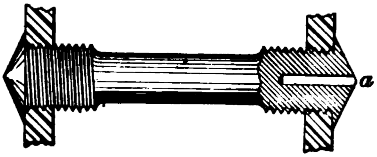


Fig. 35.

Sometimes the bolt is threaded the entire length, as in Fig. 34, or is turned off smooth in the center, as in Fig. 35. The smooth surface resists corrosion, and is less likely to fracture than the threaded bolt. Sometimes a small hole is drilled in the end, so that if the bolt breaks, the escaping steam will give warning. This is shown at *a*, Fig. 34. These bolts are $\frac{7}{8}$ inch or 1 inch in diameter.

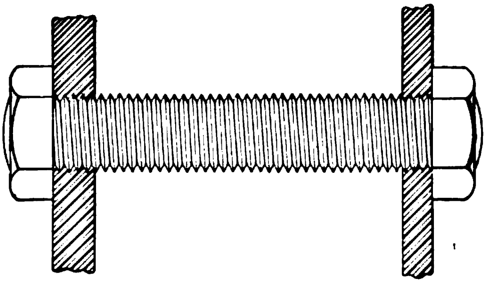


Fig. 36.

The strains which come on a stay bolt are not the same as those on rivets or on ordinary stay rods; as a matter of fact, stay bolts fail by a bending stress, and generally fracture just inside the outside sheet, due to the unequal expansion between combustion chamber or furnace and the outside boiler shell. Owing to this difference of expansion, flexible stay bolts have been designed, but have not come into general use, nor are they likely to, as they occupy considerable space and are much more complicated than the simple stay bolt. Stay bolts are made from the best quality of refined iron, which has been found to stand the strains of alternate heating and cooling better than mild steel. Iron stay bolts are more durable, because of the fibrous nature.

It should be added that boiler heads are further stiffened by channel bars or angles placed along the line of holes for the through stay rods.

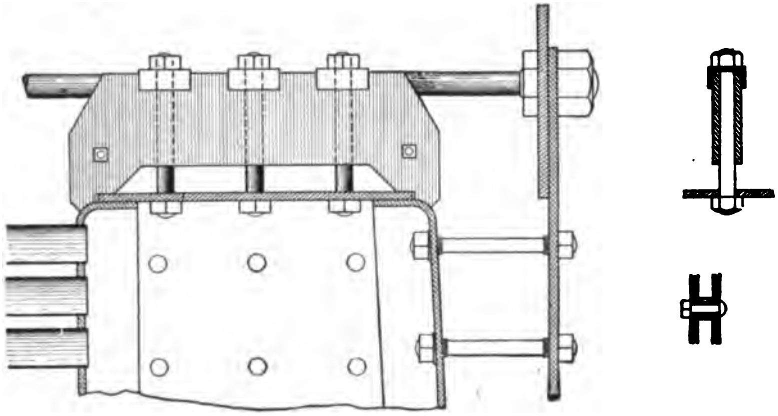


Fig. 37.

The crown sheets of fire-boxes and tops of combustion chambers are usually stayed by crown bars, which extend across the flat surfaces, as shown in Fig. 37, the ends resting on the side plates. Bolts about 4 inches apart connect the crown sheet to this girder. The girder may be a solid bar, or it may be made up of two flat plates bolted or riveted together, as shown in the figure, the stay bolts being placed between the plates at intervals

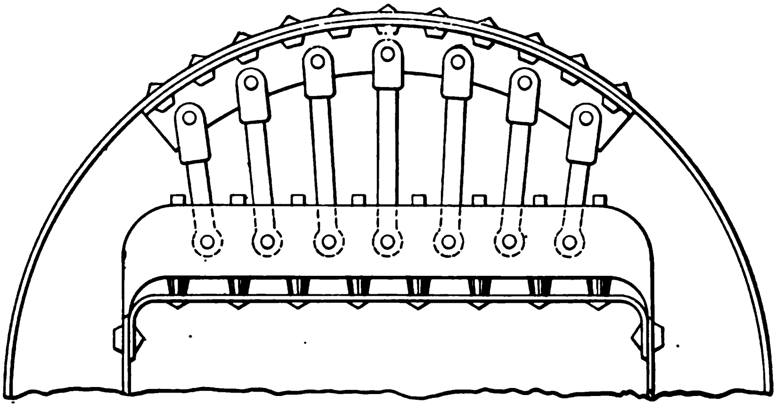


Fig. 38.

of about 4 inches. Either bolts or rivets may be used to keep the plates which form the girder from spreading. Projections are sometimes forged on the bottom of the girder, so that the stay bolts may be screwed up tightly without bending the plate.

The depth of the plates which make up the girder vary from

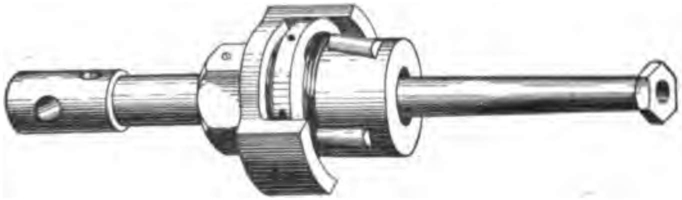


Fig. 39.

4 to 6 inches. They are from $\frac{5}{8}$ to $\frac{3}{4}$ inch in thickness. If bolts $\frac{7}{8}$ inch in diameter are used, the distance between the plates is usually 1 inch, but if larger bolts 1 inch in diameter are used, the distance should be $1\frac{1}{8}$ inches. The ends of the bars which rest upon the side plates should be carefully fitted to make a good bearing, and the area should be sufficient to prevent crushing of

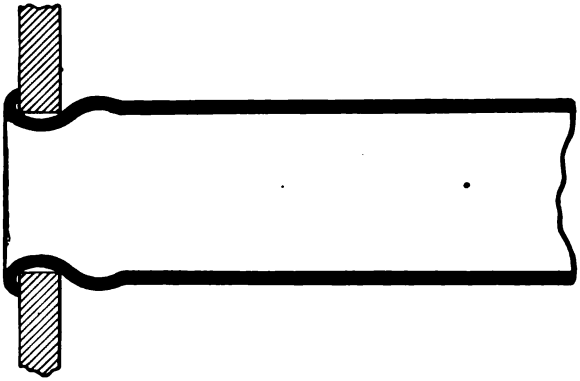


Fig. 40.

the end plates. The distance between the crown sheet and the girder should be at least $1\frac{1}{2}$ inches, so that there will be good circulation and the plates may be readily cleaned.

In some cases the girder is supported from the shell by sling stays, as shown in Fig. 38. The sling stays are connected to the girder and to an angle iron, or T-iron, which is riveted to the shell. The angle iron stiffens the shell. In designing this form of stay it is usual to make the girder strong enough to support the crown sheet without any sling stays, and these stays are used for additional support.

TUBES.

Boiler tubes are made of steel or wrought iron, but most commonly of charcoal iron and lap welded. In the formation of the lap the plate is upset, then bent around until the thickened edges lap sufficiently. It is then heated successively about 8 inches at a time, and welded over a mandrel, which is a cast-iron arm with a slightly convex top, over which the tube is placed Tubes are measured by their outside diameters, and are usually true to gauge, so that holes for them may be bored without taking measurements from the tubes themselves.

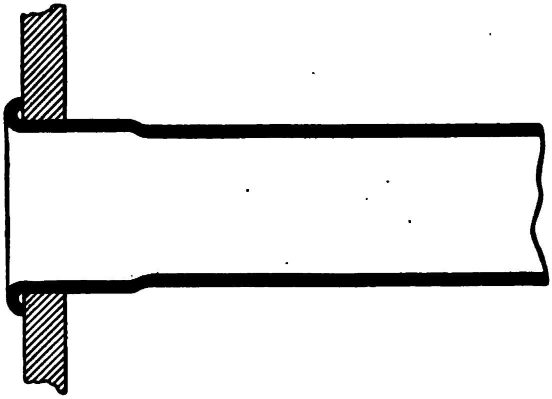


Fig. 41.

The holes for the tubes in the tube sheet are usually made in one of two ways. One method is to punch the tube holes the proper size by means of a helical punch. With this punch the metal is cut away by a shearing cut. The holes ought to be punched a little under size, and then reamed out, so that the surface against which the tubes are expanded may be good. The other method is to punch or drill a small hole at the point marking the center of the tube-hole. A drill with a post in the center, which fits the small hole, then drills the desired size of hole.

Ordinary tubes are fastened to the end plates by expanding the metal of the tube against the tube plate. This is done by a tool called an expander, of which there are two common forms. One form consists of a steel taper pin and a number of steel segments, held in place by a spring. The outside of the segments have the form to be given to the expanded tube, and the inside is a straight hollow cone, into which the steel taper pin fits. The segments are forced apart by hammering on the steel pin. In order that the metal of the tube may not be injured, the hammering should be done gradually and carefully, and the expander turned frequently. Another form, shown in Fig. 39, has a set of rolls that are forced against the inside of the tube by driving in the taper pin. The pin and rolls rotate as the pin is driven, and the rolls gradually expand the tube against the tube plate.

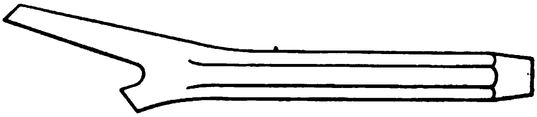


Fig. 42.

Two forms of tube expansion are shown in Figs. 40 and 41. That shown in Fig. 41 is preferable to that in Fig. 40, as the latter bears at the corners only, while the former bears against the entire thickness of the tube sheet.

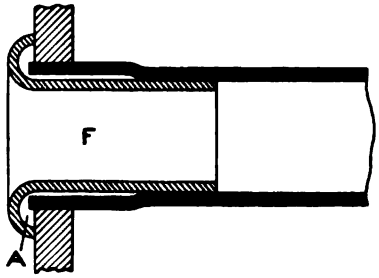


Fig. 43.

After the tubes are expanded, the ends are beaded over, as shown in Figs. 40 and 41. This adds to the strength of the connection between the tube and tube sheet. The tool commonly used for this beading is shown in Fig. 42.

Ferrules are often placed in the ends of fire tubes, and serve to protect the ends from the intense heat of the fire. The arrangement is shown in Fig. 43, the ferrule F being placed within the tube for a short distance. The space A is merely an air space.

Stay tubes are not used as extensively at the present time as they were formerly. They were very common at a time when the holding power of expanded tubes had been experimented on but little. It is now apparent from such tests that the holding power of tubes expanded, as shown in Fig. 40, is more than equal to the pressure on the spaces between the tubes of an ordinary tube plate. Stay tubes are simply heavier tubes, with the ends projecting beyond the tube sheet and threaded for shallow nuts. The ends of the tubes are frequently upset or thickened, and screwed into the tube sheet as well. This form is shown in Fig. 44.

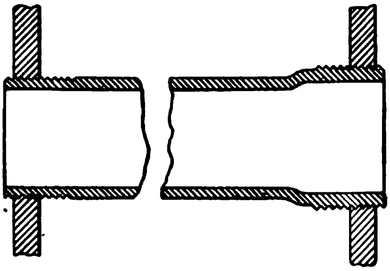


Fig. 44.

FURNACE FLUES.

Flues which are subjected to external pressure should always be cylindrical. Fig. 45 shows the section of the Adamson flue. This was an improvement over the plain furnace, as it is more elastic and allows expansion; the flanged rings also strengthen and stiffen it against collapse. The methods of building furnaces shown in Figs. 46 and 47 are not considered as good as the Adamson arrangement. Fig. 46 is too rigid, and does not

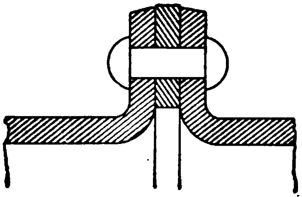


Fig. 45.

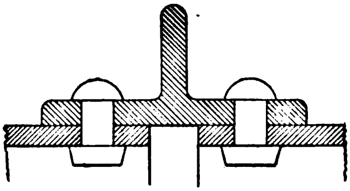


Fig. 46.

allow a free expansion and contraction. Fig. 47, on the other hand, permits of such extremely well, but both have the fault of exposing a double thickness of plates and two rows of rivets to the fire.

The corrugated flue shown in Fig. 48 is popular and, furthermore, is excellent. There is freedom for expansion throughout its whole length, thereby reducing the strains on the boiler.

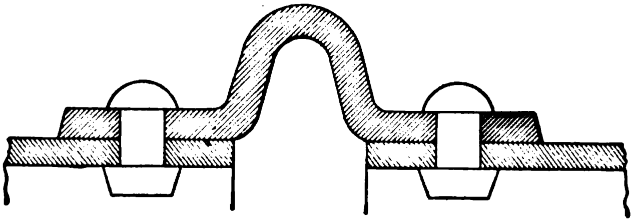


Fig. 47.

The plates should be thick enough to prevent sagging in the middle, the thickness usually varying from $\frac{5}{16}$ inch to $\frac{5}{8}$ inch. Corrugated furnaces are riveted to the rear tube sheet in the return tube boiler of the marine type, the end of the furnace being flanged at the front; and the head of the boiler is flanged around the opening cut for the furnace, which fits well into the flange.

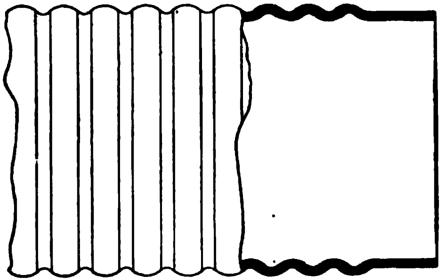
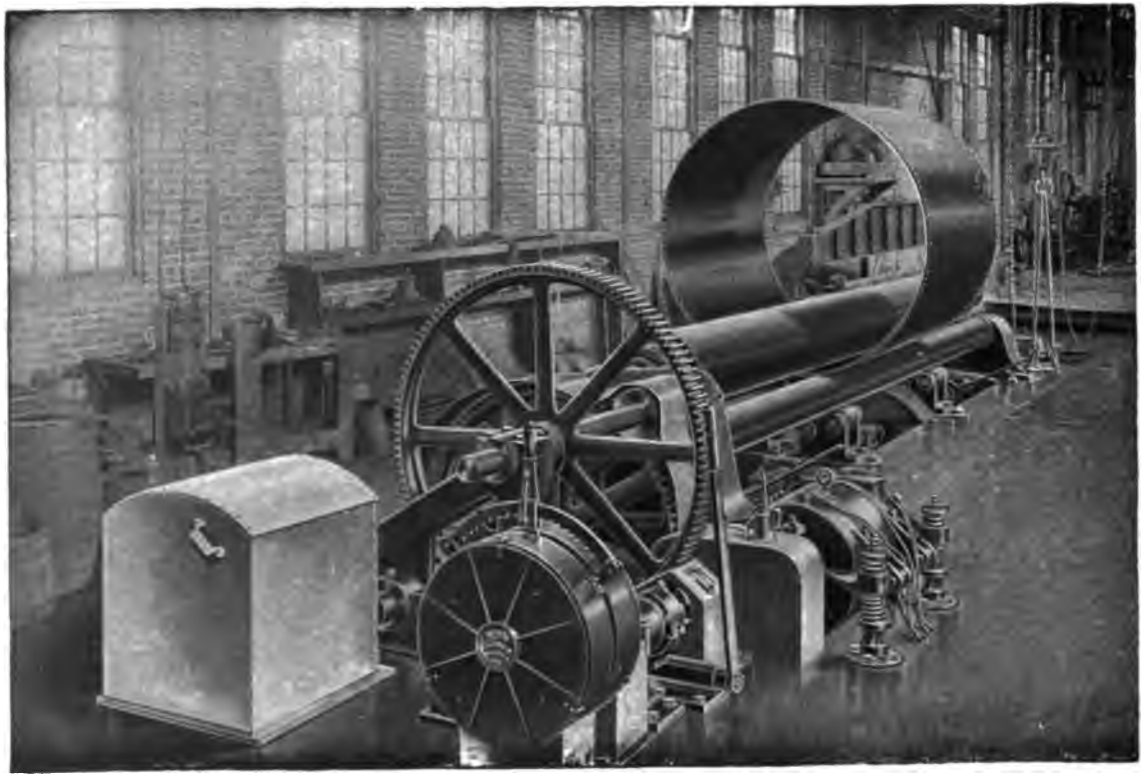


Fig. 48.

CALKING.

In order that riveted joints of boilers may be steam and water tight, they generally require calking. This process upsets the metal of the overlapping plate. or burrs down the edge,



TWENTY-FOOT BENDING ROLLS. CAHALL FACTORY.

forcing it into close contact with the lower plate, and rendering the joint steam tight.

The calking tool is similar to a chisel, the end having a variety of shapes. Fig. 49 shows a round-nosed tool which burrs down the upper plate without cutting the under plate; but it is hard to start, and in calking with such a tool the edge is first started with a sharper round-nosed tool, and then finished with one as indicated in the figure. If a square-end tool is used, as shown in Fig. 50, the under plate is likely to be cut, and the plates between the edge and the rivet be separated. The most common form of calking tool is one similar to the one shown in Fig. 49, except that the end is flat, with a slight bevel, and not round.

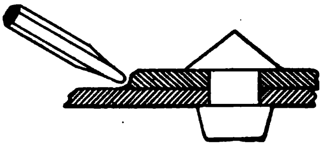


Fig. 49.

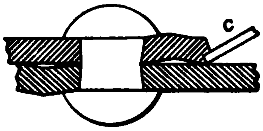


Fig. 50.

A slight bevel given the plates makes both calking and fullering more easily done. When the calking tool is thin it is sometimes driven by careless workmen into the joint, wedging the plates open. Severe and careless calking is very injurious to boilers. On the inside it often causes grooving and fracture, and the fracture of plates then follows the line of calking rather than the line of rivet holes. A pneumatic calking machine is often used in boiler shops, as it does this work about four times as rapidly as it can be done by hand. It resembles a rock drill in general principles. Air is supplied through a flexible tube, at a pressure of about 70 pounds per square inch. It makes about 1,500 strokes a minute.

BOILER DESIGN.

The rules of boiler design are controlled by practical considerations and theory, and are learned by the designer by practice only. The rules vary from place to place, and from time to time, due to progress in engineering.

The rules, methods and cautions taken up here are general, and with necessary modifications can be applied to all the more common types.

In designing a steam boiler there are several considerations that must be kept in mind. Among the most important are strength, durability, capacity to furnish the required amount of steam, convenience for cleaning, repairing and inspection, simplicity in detail, and economy both of running and first cost.

The kind, or type, to be used depends upon the work to be done, the dryness of the steam, the locality, the available space and preference of the owner. The work to be done is determined by the number and kind of engines, the constancy with which they run and the pressure. In choosing a boiler for any locality, the purity of the water, the kind of fuel and the laws which govern inspection and allowable working stress must be considered. The available space greatly influences the type and sometimes prevents choice. For instance, locomotive and marine boilers must be put in a small space. For land boilers if the floor area is limited, but there is ample height, some type of vertical boiler must be chosen.

HORSE POWER.

The unit of horse-power as decided by the American Society of Mechanical Engineers is equal to 33,305 B. T. U. From the standard steam tables in treatises on Thermo-dynamics we find that 966 B. T. U. are required to evaporate one pound of water from and at 212° F. Therefore 1 H. P. is equal to the evaporation of $33{,}305 \div 966 = 34\frac{1}{2}$ pounds of water from and at 212° F. This is also equal to the evaporation of 30 pounds of water, at 100° F. into steam at 70 pounds gauge pressure.

The first thing to do is to chose the type of boiler we are to use. Then we find how many pounds of steam are to be supplied per hour; this is found by multiplying the desired horse-power by

34½ or multiplying the horse-power of the engine or engines by the steam consumption per horse-power per hour. This is known approximately for every type of engine.

GENERAL REQUIREMENTS.

When we know these facts we design our boiler so as to have:

1. Sufficient area of grate to burn the required amount of fuel under the given draft.
2. Enough heating surface to absorb the heat of combustion.
3. Combustion chamber and flue area large enough to completely burn and carry off the products of combustion.
4. Water space sufficiently large so that a sudden demand will not cause too great a variation in water level.
5. Surface of water large compared to volume, in order that steam may be rapidly disengaged.
6. Steam space large enough to supply an irregular demand without causing a great change of pressure.
7. Steam outlet large enough to supply steam to the engine without wire-drawing.

If the outlet is not sufficiently large to supply plenty of steam, the demand will be greater than the supply and the steam will be throttled or wire-drawn, that is, it will lose some pressure.

For all common types of boilers, the proportions between the above requisites have been determined by experiment and mathematics. These relations, with simple calculations and good judgment on the part of the designer, are all that are needed for this work.

AREA OF GRATE.

A square foot of grate area will burn different weights of fuel in a given time, according to the nature of the draft. If the boiler can be made of any size, as is the case with many land boilers, a slow rate of combustion with natural draft is used, as it is the most economical. The length of the grate is limited by the distance to which a fireman can throw coal accurately. Usually 6 or 7 feet is the limit. In locomotive, torpedo boat and in some vertical land boilers, the size of grate is limited; in order to get the necessary work from the boiler, forced draft is used and the rate of combustion increases to over 100 pounds per square foot per hour. In Lancashire boilers, with two internal flues, the breadth is limited. The rate of combustion is stated in pounds per square foot of grate area per hour, and varies with the type of boiler and the draft. The following table gives the rates of combustion.

CHIMNEY DRAFT.

Cornish boilers, slow rate	4—6 lbs. per sq. ft. per hour.
Cornish boilers, ordinary rate	10—15 lbs. per sq. ft. per hour.
Factory boilers, ordinary rate	12—18 lbs. per sq. ft. per hour.
Anthracite coal, quick rate	15—20 lbs. per sq. ft. per hour.
Bituminous coal, quick rate	20—30 lbs. per sq. ft. per hour.
Marine boilers, ordinary rate,	15—25 lbs. per sq. ft. per hour.
Water tube boilers	10—25 lbs. per sq. ft. per hour.

FORCED DRAFT.

Marine boilers,	60—130 lbs. per sq. ft. per hour.
Locomotive boilers,	40—120 lbs. per sq. ft. per hour.

The evaporation per square foot of grate surface depends upon the type, the rate of combustion, condition of boiler and care in firing. The highest rate is obtained with slow rate of combustion, care and skill in firing, and clean plates and tubes. The table gives the equivalent evaporation per pound of coal for several types.

Plain cylindrical	5-- 8 pounds.
Vertical	7—10 pounds.
Cornish	6—11 pounds.
Lancashire	$6\frac{1}{2}$–12 pounds.
Galloway	9—$12\frac{1}{2}$ pounds.
Multitubular	8—12 pounds.
Water tube	6 –12 pounds.
Marine return tube	7—12 pounds.
Locomotive	6—12 pounds.

Experiment shows that an increase in the amount of coal burned per square foot of grate per hour gives an increase in the amount of water evaporated; but a decrease in amount evaporated per pound of fuel, or a decrease in economy.

To find the area of grate for a boiler. Let G = area of grate in square feet, R = rate of combustion in pounds per square foot per hour, E = evaporation per pound of coal.

$$\text{Then } G = \frac{\text{Pounds of water evaporated}}{E \times R}$$

Let us take an example. Suppose we have an externally fired multitubular boiler; assume the rate of combustion to be 12 pounds, and that our type of boiler will evaporate 9 pounds of water per pound of coal. How large must the grate be, if 2400 pounds of water are evaporated per hour?

$$G = \frac{2400}{E \times R} = \frac{2400}{9 \times 12} = 22.2 \text{ square feet.}$$

Then 22.2 square feet of grate surface are necessary. In this case the grate probably would be made 6 feet by 4 feet or 24 square feet.

TUBES.

On account of the small number of successful experiments concerning flues and chimneys, it is usual to proportion tubes, flues and chimneys, by comparison with those that have given good results. If the tubes are too large the hot gases in the centre pass up the chimney at high temperature. Now we will find the number of tubes. Let A = total area in square feet through which the smoke passes, that is, the combined internal area of all the tubes. The total area of the tubes, A, is usually made $\frac{1}{6}$ to $\frac{1}{8}$ the area of the grate. If we design our boiler to have the ratio 1 : 8 we probably will have enough area. Let us assume our tubes to be 3 inches in diameter and 16 feet long. From the table, on page 40, of lap welded boiler tubes we find that the internal area of a 3 inch tube is 6.08 square inches, the internal circumference is 8.74 inches, external circumference is 9.42 inches, and the external area is 7.07 square inches. As $\frac{1}{8}$ of our grate surface is $\frac{24}{8}$ or 3 square feet, or 432 square inches, the number of tubes will be 432 ÷ 6.08 = 71.

LAP WELDED BOILER TUBES.

External Diameter. Inches.	Internal Diameter. Inches.	Thickness. Inches.	Internal Circumference. Inches.	External Circumference. Inches.	Internal Area. Square Inches.	External Area. Square Inches.	Length of tube per sq. ft. inside. Feet.	Length of tube per sq. ft. outside. Feet.	Weight per foot. Lbs.
1	.856	.072	2.689	3.142	.575	.785	4.460	3.819	.708
1¼	1.106	.072	3.474	3.927	.960	1.227	3.455	3.056	.900
1½	1.334	.083	4.191	4.712	1.396	1.767	2.863	2.547	1.25
1¾	1.560	.095	4.901	5.498	1.911	2.405	2.448	2.183	1.665
2	1.804	.098	5.667	6.283	2.556	3.142	2.118	1.909	1.981
2¼	2.054	.098	6.484	7.069	3.314	3.976	1.850	1.698	2.238
2½	2.283	.109	7.172	7.854	4.094	4.909	1.673	1.528	2.755
2¾	2.533	.109	7.957	8.639	5.039	5.940	1.508	1.390	3.045
3	2.783	.109	8.743	9.425	6.083	7.069	1.373	1.273	3.333
3¼	3.012	.119	9.462	10.210	7.125	8.296	1.268	1.175	3.958
3½	3.262	.119	10.248	10.995	8.357	9.621	1.171	1.091	4.272
3¾	3.512	.119	11.033	11.781	9.687	11.045	1.088	1.018	4.590
4	3.741	.130	11.753	12.566	10.992	12.566	1.023	.955	5.32
4½	4.241	.130	13.323	14.137	14.126	15.904	.901	.849	6.01
5	4.720	.140	14.818	15.708	17.497	19.635	.809	.764	7.226
6	5.699	.151	17.904	18.849	25.509	28.274	.670	.637	9.346
8	7.636	.182	23.989	25.132	45.795	50.265	.500	.478	15.109
10	9.573	.214	30.074	31.416	71.975	78.540	.399	.382	22.190
12	11.542	.229	36.260	37.699	103.749	113.097	.330	.318	28.516
16	15.458	.271	48.562	50.265	187.667	201.062	.247	.238	45.200
20	19.360	.320	60.821	62.832	294.373	314.159	.197	.190	66.765

STEAM SPACE.

The steam space is frequently designed as some fraction of the volume of the shell, usually about $\frac{1}{3}$. A better way is to design it from the steam consumption of the engine. Suppose the engine uses 30 pounds of steam at 75 pounds pressure per H. P. per hour. The absolute pressure then is 90 pounds (nearly) and the specific volume at that pressure is 4.85 (from steam tables). As steam is being generated at an approximately constant rate, the supply kept on hand need not be great. If the surface for the disengagement of steam is sufficient, the ratio of the steam space to the volume of the cylinder is from 50 : 1 to 150 : 1 depending upon the speed of the engine. Experiment shows that if the steam space is equal to the volume of steam consumed by the engine in 20 seconds, it is sufficient. If the space is only equal to the steam used in 12 seconds, there may be a considerable quantity of water carried over with the steam. If the engine is slow speed, that is less than 60 revolutions per minute, the steam space should be larger.

The volume of the steam space per H. P. will be the number of pounds of steam used per H. P. in 20 seconds, multiplied by its specific volume, or $\frac{30 \times 4.85 \times 20}{60 \times 60} = .81$ cubic feet (nearly) per H. P.; and if the engine is of 75 H. P. our steam space will be $.81 \times 75 = 60.75$ cubic feet.

TUBE SPACE.

The space occupied by the tubes is equal to their volumes. The volume of one tube is its external area multiplied by the length in inches. The total volume, in cubic inches, is the above result multiplied by the number of tubes. This is reduced to cubic feet by dividing by 1728. The space occupied by the tubes will be

$$\frac{71 \times 7.07 \times 16 \times 12}{1728} = 55.77 \text{ cubic feet.}$$

WATER SPACE.

Then if we assume our steam space to be $\frac{1}{3}$ the volume of the available space in the shell, the water space will be twice the steam space, or $2 \times 60.75 = 121.5$ cubic feet.

DIMENSIONS OF BOILER.

The volume of the boiler will be:

Steam space $.81 \times 75 =$ 60.75 cubic feet.

Tube space $\dfrac{71 \times 7.07 \times 16 \times 12}{1728} =$ 55.77 cubic feet.

Water space $.81 \times 75 \times 2 =$ 121.5 cubic feet.

Total space 238.02 cubic feet.

Since the tubes are 16 feet long the area of the end will be

$$\frac{238.02}{16} = 14.87 \text{ square feet.}$$

This area gives a diameter of about $4\frac{1}{3}$ feet or 52 inches. We will make the boiler $4\frac{1}{2}$ feet or 54 inches in diameter. Then the boiler will be 16 feet long and 54 inches in diameter; with 71 tubes 3 inches in diameter. For moderate power, a common rule is to make the length about $3\frac{1}{2}$ times the diameter; by this rule our boiler is 3.55 times the diameter.

HEATING SURFACE.

The portion of a boiler that is exposed to the flames and hot gases is called the heating surface. This is made up of the portions of the shell below the brickwork, the exposed ends, and the internal surface of the tubes. If the boiler is of the water tube type, the exterior surface of the tubes is taken in place of the interior surface.

If our boiler is an ordinary multitubular boiler we can assume the heating surface to be the total inside area of the tubes plus one-half the area of the shell. Then:

Heating surface of tubes $\dfrac{8.74 \times 71 \times 16}{12} =$ 827.38 square feet.

Heating surface of shell $\dfrac{14.137 \times 16}{2} =$ 113.10 square feet.

940.48 square feet.

The ratio of heating surface to grate surface will be

$$\frac{940.48}{24} = 39.2 \text{ or about } 39.$$

As this ratio is high enough we will not alter our figures. If the ratio had been too low we could have added more tubes and found a new boiler diameter. The heating surface should not be less than 1 square yard or 9 square feet per horse-power. So 940.48 ÷ 75 = 12.54 or our boiler has 12.54 square feet of heating surface per horse-power. This is of course abundantly sufficient.

The capacity of heating surface to transmit heat to water depends upon conductivity, position of surface and temperature of furnace. In designing it is safe to follow proportions of heating surface to grate area in the various types, which experience has shown to give the best results. The following are the proportions for a few types.

Kind of Boiler.	Ratio of Heating Surface to Grate Surface.
Marine, Return tube,	25—38
Lancashire boiler,	26—33
Cornish,	27—32
Horizontal, internally fired,	40—50
Water tube,	34—65
Locomotive boiler (forced draft),	30—34
Marine,	28—32

RATIO OF GRATE SURFACE TO HORSE-POWER.

The ratio of grate surface to horse-power varies with the type, as is shown below.

Kind of Boiler.	Ratio.
Plain cylindrical,	.5 to .7
Multitubular,	.4 to .6
Vertical,	.6 to .7
Water tube,	.3
Lancashire,	.1 to .165
Marine return tube,	.12
Locomotives,	.02 to .06

Makers of boilers sometimes estimate the H. P. by the heating surface. That is the horse-power is a fraction of the heating surface. The ratio of heating surface to H. P. for several types is as follows:

Plain cylindrical,	6 —10
Multitubular,	14 —18
Vertical,	15 —20
Water tube,	10 —12
Marine return tube,	3.25— 4
Lancashire,	2.75— 4.25
Locomotive,	1 — 2

It is evident that some portions of the heating surface of a boiler have greater efficiencies than others. For instance, more heat will pass through the crown sheet as it is nearer the fire than through the last few feet of the tubes. Taking the efficiency of the crown sheet as 1, an estimate of the percentage of the other parts of a boiler is as follows:

Crown of furnace in flue,	.95
Plates of cylindrical boiler over furnace,	.90
Fire box tube plate of locomotive boiler,	.80
Water tube surface facing fire,	.70
Vertical side of fire box,	.50

If a cylindrical multitubular boiler is divided into equal sections, the section nearest the fire will evaporate more water than the one at the other end, as the gases have a higher temperature at the first section. Suppose we divide the boiler into six sections of equal length, and call the total evaporation 100 per cent. Then the per cent of evaporation per section will be approximately as follows:

Section	1	2	3	4	5	6
Evaporation	47	23	14	8	5	3

If the length of a boiler is increased another section, the evaporation will be increased a little but at the same time the radiating surface is increased. In case the addition of a section for evaporation causes a loss by radiation nearly equal to the gain in evaporation, it is not economical to add the section on account of the extra cost of the boiler. If forced draft or an increase of air of dilution is used, the boiler should be made longer to avoid waste. The air of dilution is the amount of air above that which is necessary to burn the coal.

WATER LEVEL.

If the steam space in a multitubular boiler is known the water level can be found, for the section of the steam space is a segment of a circle. In the above boiler the required steam space is 60.75 cubic feet; hence the segmental area is 60.75÷16 or 3.8 square feet, or 547.2 square inches. The height of this segment is 15.55 inches. This height is found either by calculation or from a table of segments. Then the mean water level is 15.55 inches from the top portion of the shell. The variation of water level in a boiler of this type and size should not exceed 6 inches.

END PLATE.

The end plate or tube sheet is usually made $\frac{1}{16}$ or $\frac{1}{8}$ inch thicker than the shell plates. This is done for additional stiffness, and increase of strength; the plate being weakened by drilling the holes for the ends of the tubes.

The tubes should be arranged in vertical and horizontal rows, if possible, in order that the rising bubbles of steam may not be hindered. To get good circulation the horizontal spaces should be a little greater than the vertical, and a central circulating space should be provided, if the necessary number of tubes can be put in without using the entire space. The tubes should be from $\frac{3}{4}$ to 1 inch apart, and to prevent burning of the tubes, the top row at least 3 inches below the water level, and the bottom tubes 6 inches from the shell. At this point, a drawing of the end plate should be made, to show the arrangement of tubes, etc. If it is impossible to put in the required number of tubes, without raising the water level, the diameter of the boiler must be increased. If we wish to increase the heating surface without increasing the diameter we can use smaller tubes or make the boiler a little longer.

STRENGTH OF BOILERS.

According to Pascal's Law, liquids and gases exert pressure equally in all directions. Steam in a boiler exerts the same pressure on all portions of the shell. As the pressure inside a boiler is considerably greater than that outside (the atmospheric pressure), there is a tendency to burst the shell. This tendency is resisted by the plates of the boiler.

A sphere is the strongest form to resist pressure, for since pressure is equal in all directions, there is a tendency towards enlarging the sphere and not to rupture. But a sphere has the smallest area for a given volume and, as a large heating surface is desirable, and on account of mechanical difficulties, a spherical boiler is never used. The boiler is made cylindrical to obtain greater heating surface and the loss in strength is made up by staying.

In the consideration of the strength of cylinders it is usual to divide the rupturing strains into two classes; those which tend to rupture the cylinder longitudinally and those which tend to rupture it circumferentially or transversely.

Let us examine them separately. The tendency to cause longitudinal rupture or to rend the cylinder in lines parallel with the axis, may be considered as the pressure exerted on a semi-circumference, and tending to rupture the cylinder in a plane through the diameter. Since pressure acts equally in all directions, the whole amount exerted on a semi-circumference is not exerted directly upwards and downwards. But all these forces may be resolved into their vertical and horizontal components. If we take the plane as horizontal, it is evident that the horizontal components have no tensional effect at the points of rupture. By taking the vertical components at an infinite number of points it can be proved that their sum is equal to the full pressure exerted on a rectangular plane equal to the projection of the cylindrical surface. In this case the projection is the plane through the diameter and has an area equal to the product of the length of the cylinder multiplied by the diameter of the cylinder. Then the force tending to rupture would be the pressure per square inch multiplied by the area. Let p = pressure in pounds per square inch, D = diameter of boiler, t = thickness of plate, L = length of boiler, S = tensile strength, E = efficiency of joint, and f = factor of safety. The force tending to rupture longitudinally will be, pLD. The strength of the cylinder to resist this rupturing force is represented by the tensile strength of the material multiplied by the areas of sections of metal. Or expressed algebraically is 2tLS. When rupture is about to take place the rupturing force and the strength are equal, or

$$pDL = 2tLS \quad \text{or} \quad pD = 2tS$$

from which $p = \frac{2\,t\,S}{D}$ and $t = \frac{pD}{2S}$ which are the formulas for pressure and thickness and for longitudinal strength.

The extra pressure due to increased length is balanced by the increase of metal as is shown by the elimination of the factor L of the equation.

The tendency to rupture circumferentially is evidently represented by the area of the end or $\frac{\pi D^2}{4}$ multiplied by the pressure per square inch. The strength to resist this force is the area of metal to be ruptured multiplied by the tensile strength or πDtS

$$\frac{\pi D^2}{4} \times p = \pi DtS$$

$$Dp = 4tS$$

By comparing these two formulas we see that with the same internal pressure, diameter and thickness of shell, a cylindrical boiler is twice as strong transversely as it is longitudinally, hence the greatest tendency to rupture is along the longitudinal seams.

Therefore, in designing the thickness of shell we use the formula for longitudinal rupture,

$$pD = 2tS \text{ or } t = \frac{pD}{2S}$$

or, inserting the factors for efficiency of joint and factor of safety,

$$pD = \frac{2tSE}{f}$$

$$\text{For allowable pressure} \quad p = \frac{2tSE}{fD}$$

$$\text{For thickness of shell} \quad t = \frac{fDp}{2SE}$$

Now let us find the thickness of the boiler that we are designing. Suppose after testing our material we find that its ultimate tensile strength is 54,000 pounds per square inch. In this case 6 will be sufficiently large for a factor of safety. This factor can be reduced if the efficiency of the joint is large. Let us assume that our joint has an efficiency of 70%. This is merely a supposition because we have not yet constructed the joint; but we assume a factor in order to find a trial thickness.

$$\text{Then } t = \frac{fDp}{2SE} = \frac{6 \times 54 \times 75}{2 \times 54000 \times .7} = .32 \text{ or about } \tfrac{5}{16} \text{ inches.}$$

RIVETED JOINTS.

The best knowledge of the strength and proportions of riveted joints can be obtained by tests of full sized pieces. Let us consider the strength and efficiency mathematically. Riveted joints may fail in several ways. 1. By shearing the rivets. 2. By tearing the plate at the reduced section between the rivets. 3. By crushing the plate or rivets where they are in contact. 4. By cracking the plate between the rivet hole and the edge of the plate. As the lap in practice can always be made sufficiently wide a joint need never fail in this last way.

As all stresses may be resolved into the three kinds, tensile, compressive and shearing, we will investigate for these stresses. Let P = the tensile stress transmitted from one plate to the other by a single rivet, t = the thickness of the plate, d = the diameter of the rivet, p = the pitch, and S_t, S_s and S_c the unit stresses in tension, shear and compression respectively produced by P on the plates and rivets. Therefore the tension on the plate, P will be equal to the area of the metal between the rivets multiplied by its unit tensile stress, or

$$P = t\,(p - d)\,S_t$$

For shear, P will equal the area of the rivet multiplied by the unit shearing stress, or

$$P = \tfrac{1}{4}\,\pi\,d^2S_s$$

For compression, the stress is supposed to be equivalent to a stress uniformly distributed over the projection of the cylindrical surface on a plane through the axis of the rivet. Then P will be equal to the area of the projection multiplied by the unit compressive stress, or

$$P = tdS_c$$

The above formulas are for single riveted lap joints. If another row of rivets is used the plates should have a wider lap. Let $p =$ the pitch in one row; then the stress will be distributed over two rivets.

The three formulas in this case will be.

$$\begin{aligned} P &= t(p - d)S_t \\ P &= 2 \times \tfrac{1}{4}\pi d^2S_s \\ P &= 2\,tdS_c \end{aligned}$$

For single riveted butt joint, the shear comes on two rivets; this is called double shear. The above formulas become

$$\begin{aligned} P &= t(p - d)S_t \\ P &= 2 \times \tfrac{1}{4}\pi d^2S_s \\ P &= tdS_c \end{aligned}$$

The efficiency of a joint is the ratio of its allowable stress to the allowable stress of the uncut plate. The allowable stress of the plate is represented by the formula ptS_t.

Then the efficiency for tension is, $E = \dfrac{t(p - d)S_t}{ptS_t} = \dfrac{p - d}{p}$

For shear, $E = \dfrac{\frac{1}{4}\pi d^2S_s}{ptS_t}$ or $\dfrac{\frac{1}{4}\pi d^2S_sc}{ptS_t}$

For compression, $E = \dfrac{tdS_c}{ptS_t}$ or $\dfrac{dS_c}{pS_t}$ or $\dfrac{dS_ca}{pS_t}$

In the above formulas, $a =$ the number of rivets in the width p, and $c =$ the number of rivet sections in the same space. The smallest value of E is to be taken as the efficiency of the joint.

In designing, we try to get a joint in which all parts will have equal strength or the resistance of the plate to tension will equal the resistance of the rivets to shearing and each will equal the resistance of the rivet to compression or crushing. This will be the case if the three efficiencies are equal.

Solving for d in the second and third we get

$$\frac{\frac{1}{4}\pi d^2S_sc}{ptS_t} = \frac{dS_ca}{pS_t} \text{ or } d = \frac{4aS_ct}{\pi cS_s}$$

If we know t we can find d from the above equation.

To find the pitch we make the first equation equal to the third, or the formula for tension equal that of compression, and solve for p

$$\frac{p-d}{p} = \frac{dS_c a}{pS_t},\ p = d\left(\frac{S_c a}{S_t} + 1\right)$$

substituting the value for d, obtained above,

$$p = \frac{4\ aS_c t}{\pi\ cS_s}\left(\frac{S_c a}{S_t} + 1\right)$$

To get the formula for efficiency we insert these values for d and p, in any of the formulas for efficiency already obtained. For instance:

$$E = \frac{p-d}{p} = \frac{\frac{4\ aS_c t}{\pi\ cS_s}\left(\frac{S_c a}{S_t} + 1\right) - \frac{4\ aS_c t}{\pi\ cS_s}}{\frac{4\ aS_c t}{\pi\ cS_s}\left(\frac{S_c a}{S_t} + 1\right)}$$

$$E = \frac{1}{1 + \frac{S_t}{aS_c}}$$

A good joint can be designed without these formulas (in fact they serve as a guide only), if attention is paid to the rules deduced from tests and conforming to good practice by experienced engineers and boiler makers. In designing a riveted joint, good practice favors the following:

The pitch of rivets, for single riveting, should be about 2½ times the diameter of the rivets and for double riveting about 3¾ times the diameter.

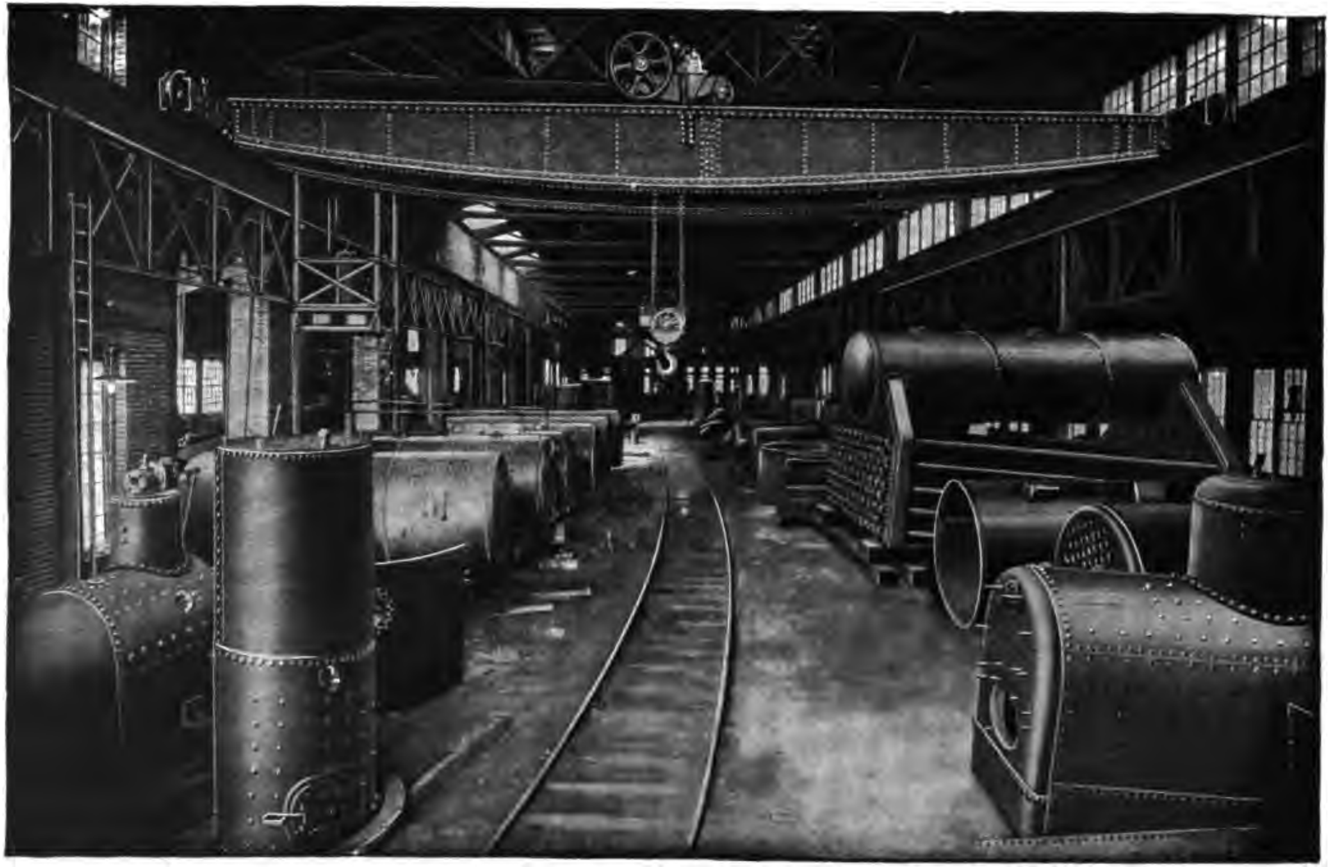
The pitch near a calked edge must not be too great for proper calking.

Rivets must not be too near together.

The lap, or the distance from the centre of the rivet to the edge of the over-lapping plate should be at least 1½ times the diameter of the rivet.

The diameter of the rivet is usually nearly twice the thickness of the plate and should never be less than the thickness of the plate.

The riveted seam must contain a whole number of rivets. and similar seams should have the same pitch.



ERECTING SHOP.
E. Keeler Company.

The distance between rows, for double riveting, is about twice the diameter of the rivets.

In double butt riveting the rivets in double shear have $1\frac{3}{4}$ times the single section instead of 2.

The thickness of double butt straps should not be less than $\frac{5}{8}$ the thickness of the plate (each); single butt straps not less than $\frac{9}{8}$.

No one set of rules can be laid down for the best pitch of rivets for all circumstances of pressure, quality of plates, etc. The following table of proportions of riveted joints gives results for average practice in boilers of up to about 150 pounds pressure.

TABLE OF LAP JOINTS.

Thickness of Plate. Inches.	Diameter of Rivet. Inches.	Diameter of Hole. Inches.	Pitch. Inches.		Efficiency.	
			Single Riveted.	Double Riveted.	Single Riveted.	Double Riveted.
$\frac{1}{4}$	$\frac{5}{8}$	$\frac{11}{16}$	2	3	.66	.77
$\frac{5}{16}$	$\frac{11}{16}$	$\frac{3}{4}$	$2\frac{1}{16}$	$3\frac{1}{8}$	.64	.76
$\frac{3}{8}$	$\frac{3}{4}$	$\frac{13}{16}$	$2\frac{1}{8}$	$3\frac{1}{4}$	.62	.75
$\frac{7}{16}$	$\frac{13}{16}$	$\frac{7}{8}$	$2\frac{3}{16}$	$3\frac{3}{8}$	.60	.74
$\frac{1}{2}$	$\frac{7}{8}$	$\frac{15}{16}$	$2\frac{1}{4}$	$3\frac{1}{2}$	.58	.73

As the stress on the transverse section is one-half that on a longitudinal section, a single or double lap joint is sufficient for any ring seam. For externally fired multitubular boilers with shell plates less than $\frac{1}{2}$ inch thick, single riveted ring seams are used. For our boiler, the plates being $\frac{5}{16}$ inch thick, we will use rivets $\frac{11}{16}$ inch in diameter, as this agrees with good practice. From the table, the pitch for a $\frac{11}{16}$ inch rivet for single riveted lap joint is $2\frac{1}{16}$. Then as our ring seam is $3.1416 \times 54 = 169.65$ inches and pitch $2\frac{1}{16}$ inches we will have 82 + rivets. But as we must have a whole number of rivets we will alter the pitch slightly and use 82 rivets with a pitch of 2.069 inches. The result depends, in each case, upon the kind of joint used in longitudinal seams. This merely shows the general method. The lap will be $\frac{11}{16} \times \frac{3}{2} = 1\frac{1}{32}$ inch.

For the longitudinal seams we will use double butt joints with single riveting. The thickness of the butt straps will be $\frac{5}{16} \times \frac{5}{8} = .20$ (nearly). To be on the safe side we will make the butt straps $\frac{1}{4}$ inch thick. The pitch for double butt joints is usually about 4 times the diameter of rivets, so in this case it will be $4 \times \frac{11}{16} = 2\frac{3}{4}$ inches. We will use the same amount of lap for this joint as for the lap joint, that is $1\frac{1}{32}$ inches.

SECTIONS.

The boiler is made up of rings or sections. The length of sections is often made equal, for convenience in ordering and cutting plates. The length is limited by the width of plate obtainable and the size of the riveting machine. This boiler being 16 feet long would probably be made in three sections, but the lengths should be so adjusted as not to bring the ring seam over the hottest part of the fire.

FLUES.

The internal pressure at which the boiler shell will rupture can be calculated; but the external pressure which will collapse a flue can be determined only by experiment. External pressure tends to increase any imperfection of shape. For instance, if a flue is slightly oval, the external pressure tends to make it more flat. The strongest form to resist external pressure is evidently the circle. When considering the strength of flues length is very important.

If a lap joint is used the flue will not be a true cylinder, for this reason welded or butt joints are preferable.

Fairbain gives the formula, $P = \frac{806{,}000\ t^{2.19}}{ld}$ for calculating the collapsing pressure of flues, l = length of flue in feet, d = diameter in inches, t = the thickness in inches, P = pressure per square inch. The exponent of t is often taken as 2 instead of 2.19 for convenience. This formula is empirical and was prepared from his experiments.

Hutton gives, $P = \frac{C\ t^2}{d\sqrt{L}}$. In which C is a constant, which is 600 for wrought iron and 660 for mild steel, L = length in inches, d = external diameter in inches, and t = thickness in *thirty-seconds* of an inch. Results by Hutton's formula agree more nearly to those by experiment than do Fairbain's.

If the flues are oval, d in the above formula = the major axis.

Flues are strengthened by putting in hoops at stated distances. These hoops are made of T iron or angle iron.

TUBES.

The materials for tubes are iron and steel. The tubes must be tough to resist cutting by cinders. If iron is used it should have a tensile strength of at least 45,000 pounds per square inch, with an elongation of 15 to 20 per cent. If steel, the elongation should not be less than 26 per cent, when tested before being rolled. If the steel welds well there need not be any limit to its tensile strength. The ends of tubes should be annealed after manufacture. The thickness of tubes is always greater than that required to prevent collapsing, in order to weld and expand in the tube sheet. It is often desirable to use part of the tubes as stays; for this purpose the tubes are made thick enough to take a shallow nut outside the tube plate.

STAYING.

As large a portion as possible of the shell of a boiler is made cylindrical, for in this form plates can be made sufficiently strong without the aid of stays or braces. But all flat surfaces must be stayed; not only to prevent rupture, but also to provide against distortion and grooving. The theoretical investigation of the strength of flat surfaces, can be worked out only with higher mathematics. From the formula deduced, the solid end plate would have to be about 2 inches thick for a boiler only 3 feet in diameter with plates $\frac{3}{8}$ inch thick. It is evident that the flat ends if of ordinary thickness must be strengthened by stays or braces. The calculation of stresses in a flat plate, supported by stays, can be calculated only when the supported points are in rows thus dividing the surface into equal squares. Even when the stays are not to be placed in rows forming squares, it is well to make the calculation for a standard.

The equation for finding the area supported by a stay rod is,

$$a^2 = \frac{9t^2S}{2p}$$

in which a^2 = the area supported, t = the thickness of the end plate, S = the allowable stress on the area of the rod, and p = the working steam pressure. Let us find the area supported in our multitubular boiler. In order to provide for future corrosion we will use a factor of safety of 12 and assume, in the absence of exact knowledge, the ultimate breaking strength of the rod to be 60,000 pounds per square inch. It is usual to make the diameter of the rods one to two inches, so we will make ours $1\frac{1}{2}$ inches in diameter with an area of 1.767 square inches. Then the stress per rod is 5,000 × 1.767 = 8,835 pounds.

$$a^2 = \frac{9t^2S}{2p} = \frac{9 \times \frac{1}{4} \times 8{,}835}{2 \times 75} = 132.5 \text{ square inches.}$$

Then as the rod supports 132.5 square inches and the segment of the steam space is 547.2 square inches, the number of rods will be $\frac{547.2}{132.5} = 4$.

The same formula will apply in finding the number of short screw stay-bolts of the fire box.

Suppose we wish to use a diagonal or crow foot stay, making an angle of 20° with the shell. If the rod is 1 inch in diameter and the stress is limited to 7,000 pounds, then it will carry a pull of .7854 × 7,000 = 5497.8 pounds, and since it makes an angle of 20°, the pull perpendicular to the head will be 5497.8 × cos. 20° = 5497.8 × .9397* = 5,166 pounds. If the end is fastened by two rivets or bolts each will carry 2,583 pounds. If each rivet or bolt supports a square with a side equal to a, then $5{,}166 = 75\,a^2$

$$a^2 = \frac{5166}{75} = 68.9 \text{ square inches (nearly).}$$

* **Note.** Taken from a table of cosines.

UPTAKE.

The area of the uptake, like the area of the tubes, is made about $\frac{1}{7}$ to $\frac{1}{8}$ of the area of the grate. We find that $\frac{1}{8}$ of the grate surface is 432 square inches. If we make the uptake 12 inches deep measured with the length of the boiler, it will be 432 ÷ 12

= 36 inches wide. The opening of the shell at the front end will be 12 inches deep and the plate cut down until it is 36 inches wide.

MANHOLES.

The manhole and handhole should be strong enough and stiff enough to sustain the stresses due to the direct steam pressure and from the stresses of the plates. The calculation of the strength of the manhole ring is difficult and the results obtained very uncertain, so they are made of forms and dimensions that have been used in good practice and given good results. These fittings are bought in steel forgings. Boiler makers design the forged rings which lie close to the shell, of a section at least equal to the section of the plate that is cut out. The bearing surfaces of the manhole cover and that of the lip against which the cover bears, should be machined to make a good smooth joint. The joints are made tight by gaskets about $\frac{3}{4}$ of an inch wide.

Hand holes are constructed similarly to manholes, and often have a taper key in place of a bolt and nut, because the nut is exposed to fire and after it has been in place some time, is often difficult to remove with a wrench.

BRACKETS.

Boilers of the multitubular types are supported by brackets usually made of cast iron. Boilers up to 16 feet long have four brackets and those more than 16 feet long have six brackets. The brackets for this boiler should be about 10 inches long, measured with the length of the boiler, and about 15 inches wide. They are riveted to the boiler with nine or ten rivets $\frac{7}{8}$ to 1 inch in diameter. The rivets can be made large, as a large rivet makes a strong joint, and in this case the pitch is not governed by calking.

The load on the brackets can be estimated by calculating the weight of the boiler full of water and adding the weight of all the parts supported by the boiler. These parts include pipes, valves, gauges, brickwork covering, etc. This load should be divided as nearly equal as possible among the four brackets, so that the tendency of the boiler toward bending shall be small.

Brackets are set above the middle line of the boiler in order that the flanges may be protected by the brickwork setting. They are usually 3 or 4 inches above the middle.

CHIMNEYS.

At the present time, the knowledge of chimneys and chimney draft is slight. The theories given are worth but little as they are based upon data which is entirely insufficient. As to the design and proportions of chimneys, there are no systematic statements and rules that can be used.

Chimneys are usually designed from empirical formulas and from tables, compiled from proportions of chimneys that have furnished sufficient draft, etc.

The draft produced in a chimney is due to the difference in temperature, and consequently difference in pressure, between the gases inside the chimney, and the air outside. The gases in the chimney being lighter rise toward the top and air rushes in at the bottom to fill the space left by the hot gases. This air as it becomes heated grows lighter and rises, thus a continuous circulation is kept up. The temperature of the gases in the chimney is considered to be about 600° F. for chimney calculation, as practice shows this to give good draft under economical conditions.

After making several assumptions, based on experiments, the following formula has been deduced:

$$H. P. = 3.33 (A - .6 \sqrt{A}) \sqrt{h}$$

in which H. P. = horse-power, A = area of the chimney, and h = the height above the grate.

The following table on page 46 has been calculated from this formula. This table is used to a considerable extent with satisfactory results.

The part of the table which is used for ordinary proportions is filled in. If proportions are taken from the table rather than from the formula, the results will give better proportions.

To find the area of the top of the chimney for a given coal consumption, the following empirical formula has been stated.

$$A = \frac{H. P. \times B \times 12}{\sqrt{h}}$$

in which A = area, H. P. = horse-power of boiler, B = number of pounds consumed per H. P. per hour and h = height of chimney in feet.

This area A is the area in square inches at the top.

Diameter, in Inches.	Height of Chimneys and Commercial Horse-power.											Side of square in inches.	Effective area, square feet.	Actual area, square feet.
	50 ft.	60 ft.	70 ft.	80 ft.	90 ft.	100 ft.	110 ft.	125 ft.	150 ft.	175 ft.	200 ft.			
18	23	25	27									16	0.97	1.77
21	35	38	41									19	1.47	2.41
24	49	54	58	62								22	2.08	3.14
27	65	72	78	83								24	2.78	3.98
30	84	92	100	107	113							27	3.58	4.91
33		115	125	133	141							30	4.48	5.94
36		141	152	161	173	182						32	5.47	7.07
39			183	196	208	219						35	6.57	8.30
42			216	231	245	258	271					38	7.76	9.62
48				311	330	348	365	389				43	10.44	12.57
54					427	449	472	503	551			48	13.51	15.90
60					536	565	593	632	692	748		54	16.98	19.64
66						694	728	776	849	918	981	59	20.83	23.76
72						835	876	934	1023	1105	1181	64	25.08	28.27
78							1038	1107	1212	1310	1400	70	29.73	33.18
84							1214	1294	1418	1531	1637	75	34.76	38.48
90								1496	1639	1770	1893	80	40.19	44.18
96									1876	2027	2167	86	46.01	50.27

Another method which is much more simple is to design the area of the chimney, as we have designed the total tube area; that is, about $\frac{1}{8}$ the grate area. This ratio for chimneys is sometimes about $\frac{1}{7}$ and decreases to $\frac{1}{9}$ and for very tall chimneys to $\frac{1}{10}$.

From the table we find the chimney to have an area at the top of about 3.98 square feet, assuming it to be 60 or 70 feet high. This area gives a diameter of 27 inches if circular, or 24 inches if square.

Let us calculate it from the formula $A = \dfrac{\text{H. P.} \times B \times 12}{\sqrt{h}}$

We must either assume or calculate B. As the calculation is very easy it would be better than any assumption. The total amount of coal burned per hour equals 12 × 24 or 288 pounds. The amount per H. P. per hour is 288 ÷ 75 or 3.84 pounds.

Then assuming the chimney to be 60 feet high,

$A = \frac{75 \times 3.84 \times 12}{\sqrt{60}} = 446$ square inches, or about 24 inches in diameter if circular and 21 inches if square.

By the last method the area of the chimney will be 24 ÷ 8 or 3 square feet, or 432 square inches, giving practically the same result as with the formula.

As the table is reliable and gives us the larger area, we will use it and be on the safe side; also as the amount of coal burned per hour by the draft in a chimney can be found by multiplying the horse-power in the above table by 5, the chimney with an area of 3.98 square feet and 60 feet high will burn $72 \times 5 = 360$ pounds of coal per hour. The boiler in question burns only 288 pounds, so the chimney is sufficiently large.

Chimneys are usually of brick or of steel plates. If of steel they are always circular. When made of brick they are circular, square or hexagonal. With a given draft area, a circular chimney requires the least material, since a circumference has the least perimeter for a given area; it also presents less resistance to wind.

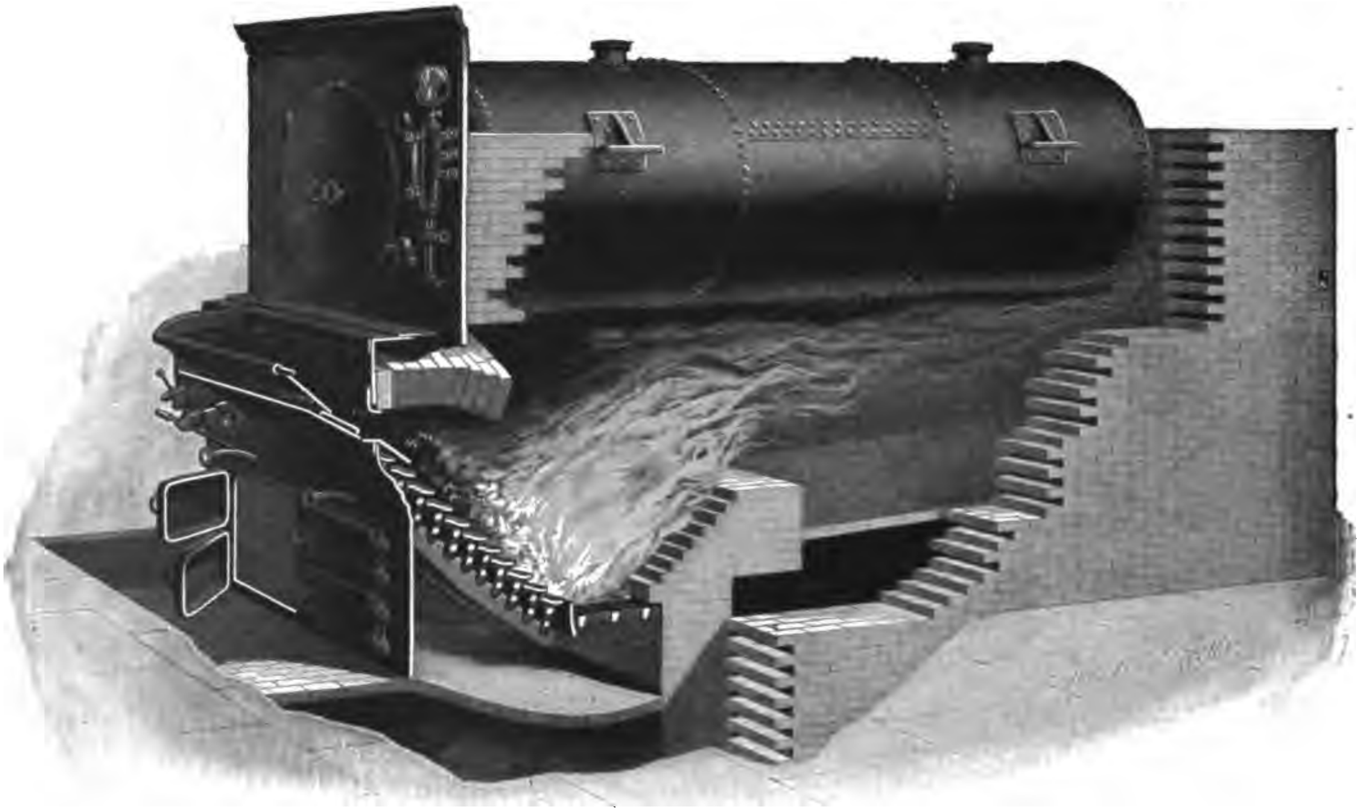
A steel chimney is made up of plates of steel riveted together. The shell is bolted through a foundation ring of cast iron to the stone foundation. It has a straight taper to the top, which is finished, for appearance with light plates. The shell is lined with fire-brick, with a thickness which varies from 12 to 18 inches at the bottom to about 2 to 4 inches at the top. This lining is used to prevent heat being lost from the shell and does not add to the strength of the chimney.

A brick chimney is built in two parts; *a* the outer shell, which resists wind pressure and *b*, the lining which is the flue. This flue is made separate from the external shell in order that it may expand, when the chimney is full of hot gases, without straining the outer shell.

The interior of both steel and brick chimneys are often cylindrical while the exterior tapers. The taper is about .3 inch to the foot. The brick at the base of the chimney is splayed out to make a large base.

As good natural earth should carry from 2000 to 4000 pounds per square foot, the base of the chimney should be large enough so that this pressure will not be exceeded.

The external shell is calculated for wind pressure and the weight of brick. This calculation for wind pressure involving higher mathematics will not be treated here. The lining is calculated for compression due to weight. The design, both of the chimney and its foundation, should be made by a competent engineer of experience, on account of disastrous results should a chimney fall.



GENERAL PERSPECTIVE OF THE RONEY MECHANICAL STOKER AS APPLIED TO A HORIZONTAL TUBULAR BOILER.

TYPES OF BOILERS.

Generally speaking, a steam boiler is a closed metallic vessel in which steam is generated from water by the application of heat. As steam is under pressure it is evident that the vessel must be strong and tight.

To operate the boiler safely and economically there must be certain fittings and accessories—some of these are used in the care of the boiler, while others serve to increase the economy. Among the most important attachments and appurtenances may be mentioned the following:

A feed pump or injector, with valves, piping, etc., to supply water to the boiler.

Gage cocks and glass water gage to show the attendant the height of water or the water level, as it is called, in the boiler.

A pressure gage to show the pressure of steam in the boiler. The pressure is usually measured in pounds per square inch.

A safety valve to allow steam to escape from the boiler when the pressure exceeds a certain fixed amount. This attachment, being a safety device, should be automatic and reliable.

A blow-off pipe, with its valves, to blow out sediment from the boiler, reduce the amount of water in the boiler, or empty it.

A steam pipe, with its valves, to conduct the steam from the boiler to the place where it is to be used.

Manholes and handholes, with covers, for examination, repairs, and cleaning.

* **Fusible plugs** to give warning when the water level becomes too low, or melt and allow the water to escape.

* **High- and low-water alarms** to give warning when the water level is too high or too low.

* **A heater** to raise the temperature of the feed water as nearly as possible to that of the water in the boiler.

***Note. Although the last three are desirable, they are not absolutely necessary, as a boiler can be successfully operated without them.**

In addition to these there are other attachments such as:

Lugs or brackets for supporting the boiler.

Masonry for setting the boiler and keeping it in position, and in many cases to keep the hot gases in contact with the shell.

Furnace fittings, including grate bars, bearer bars, dampers, fire doors, ashpit doors, etc.

The chimney to carry away the waste gases and create draft.

Tools, such as shovels, slice bars, scrapers, tube brushes, etc.

DEFINITIONS.

The following definitions should be remembered in connection with the terms used in designating the various classes.

A fire-tube boiler is one having the heating surface composed largely of tubes which are surrounded with water, the hot gases passing through them.

A water-tube boiler is also composed of tubes, but in this case *water* flows through the tubes, while the hot gases pass around and among them.

In a sectional boiler the tubes and corresponding headers form comparatively small units. Each unit is complete in itself; that is, it is in communication with a steam and water drum but is independent of the other units.

A non-sectional boiler is one having all the tubes in communication with one another; in other words, all or nearly all the tubes are expanded into a common header or drum. The boiler is not made up of units.

A single-tube boiler is made up of plain tubes.

A double-tube boiler has a small tube inside of the regular tube and concentric with it.

A boiler is **externally-fired** when the furnace is separate from the shell; in such boilers the fire is usually placed in a brick furnace.

In the **internally-fired** boiler the grate is inside of a flue which is within the shell.

A fire-box boiler is one having the fire within a fire box which, although external to the shell, is rigidly connected to it. The fire box is usually made of steel plates instead of brick as in the case of the externally-fired boiler.

CLASSIFICATIONS.

The almost endless variety of boilers now in use is due largely to the many conditions under which they are used. Other reasons for the numerous forms are the great latitude in design and construction, and the competition among engineers, who have, during the last century, sought to produce, at moderate cost, steam generators that will be safe, durable, and economical.

The necessity for careful classification before discussing the details is apparent when one considers the similarities and differences. Much valuable time may be saved by selecting some make of boiler to represent a given class. Still further, the classification reduces the chances of overlooking interesting features.

CLASSIFICATION.

According to Use.

- **Stationary**
  - Early Forms.
  - Plain Cylindrical
  - Single Flue, Externally-fired
  - Flue Boilers
    - Cornish (single-flue)
    - Lancashire (two-flue)
    - Galloway
  - Multitubular
    - Externally-fired
    - Internally-fired (return-tube)
  - Fire-box Boilers
    - Horizontal
    - Vertical
  - Water-tube Boilers
    - Straight-tube
    - Curved-tube
    - Horizontal
    - Vertical
    - Sectional
    - Non-sectional
  - Mixed Types
  - Peculiar Forms
- **Marine**
  - Early Forms (box or rectangular)
  - Scotch or Drum
  - Return-tube
  - Through-tube
  - Water-tube
    - Curved-tube
    - Straight-tube
    - Sectional
    - Non-sectional
  - Launch Boilers
- **Locomotive**
  - Multitubular fire-box (common form)
  - Wooten Type
  - Corrugated Furnace
  - Peculiar Forms

Boilers may be classified in many ways. They may be divided into the following great classes: Fire-tube and water-tube, vertical and horizontal, stationary and non-stationary, or externally-fired and internally-fired. They may also be classified according to uses or according to forms of construction. For illustration, two classifications, of which the following seems better for this discussion, are given.

CLASSIFICATION.

According to Form of Construction.

Early Forms.

- **Flue**
  - Cornish (single-flue)
  - Lancashire (two-flue)
  - Galloway
  - Single Flue (externally-fired)
- **Fire-tube** (Multitubular)
  - Horizontal (common form)
  - Vertical
  - Return-tube
  - Through-tube
  - Fire-box
  - Peculiar Forms
- **Water-tube**
  - Horizontal
    - Straight-tube
      - Sectional
      - Non-sectional
    - Curved-tube
      - Sectional
      - Non-sectional
  - Vertical
    - Straight-tube
    - Curved-tube
  - Peculiar Forms

Mixed Types.

EARLY FORMS.

The earliest boilers of which we have reliable record were spherical. They were of cast iron and set in brickwork. It was customary to set this type of boiler with the fire underneath and construct flues in the brickwork to conduct the hot gases around the boiler just below the water level. The hot gases passed entirely around the boiler before escaping to the chimney.

The Haystack Boiler. The next form to be generally used was that invented by Newcomen in 1711. On account of its peculiar shape it was called the "Haystack" or "Balloon" boiler. It was of wrought iron and had a hemispherical top and arched bottom. The fire was placed underneath the arched portion; the hot gases surrounding the lower part of the boiler. An improved form of the Haystack boiler is shown in Fig. 1. Smeaton placed the fire inside the shell and arranged internal flues for conducting the hot gases to the chimney. This arrangement increases the heating surface and consequently the economy of the boiler.

The Wagon Boiler. To still further increase the heating surface, James Watt introduced his "Wagon" boiler. This form is shown in Fig. 2. The top was cylindrical and the sides curved inward. The curved plates assisted in the formation of flues on either side. The hot gases passed from the grate, underneath the boiler to the rear, through the left-hand flue to the front, then through the right-hand flue to the rear and thence to the chimney. This was called the *wheel draft* because the gases passed entirely around the boiler. In the large sizes a flue was placed in the boiler. The products of combustion returned through this flue to the front after passing under the boiler to the rear, as in the small sizes. On issuing from the flue at the front, the gases divided and passed to the chimney at the rear by means of the flues in the brickwork. This form of draft was called the *split draft*.

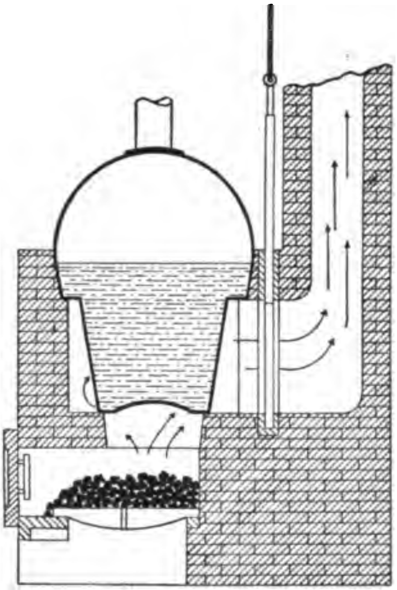


Fig. 1.

Watt used a column of water in the vertical feed pipe as a pressure gage; the rise and fall of this column also controlled the damper. The feed was regulated by a float.

MODERN BOILERS.

Although such boilers as the Haystack, Wagon, and others were fairly satisfactory in the period in which they were invented, they could not stand the higher pressures that soon became common.

About the beginning of the nineteenth century the cylindrical boiler was introduced. The earliest forms were the plain cylindrical boiler and the "Egg-end" boiler. The difference was in the form of the ends — those of the former were flat and of cast iron, while the ends of the latter were hemispherical and made of wrought iron. The egg-end boiler required no staying or bracing because its form is, with the exception of a sphere, the strongest to resist internal pressure.

The Cylindrical Boiler consisted of a shell of wrought-iron boiler plate and ends of the same material or of cast iron. It was set in brickwork as shown in Fig. 3. The boiler was about two-thirds filled with water, the remaining third forming the steam space. To collect and store the steam as it rose from the water a steam dome was added. The steam pipe was attached to the dome to which the safety valve also was connected. The hot gases from the fire passed under the boiler to the rear and then to the chimney.

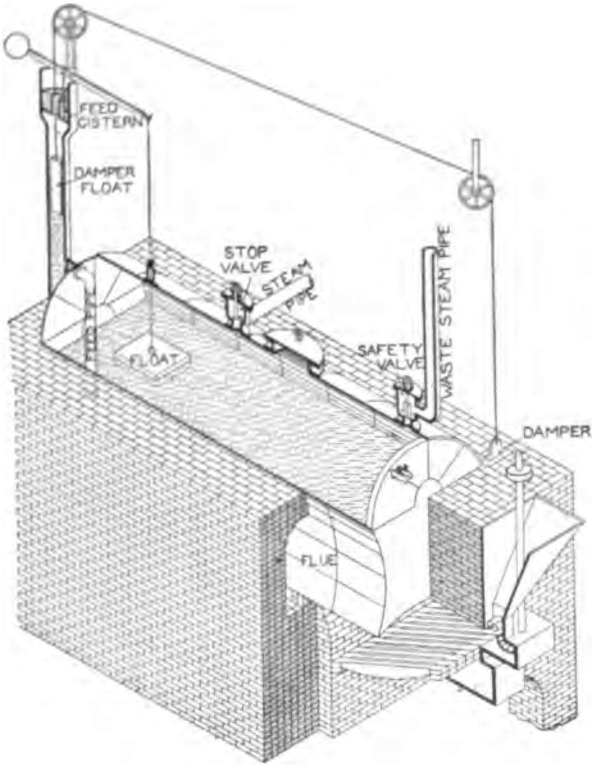
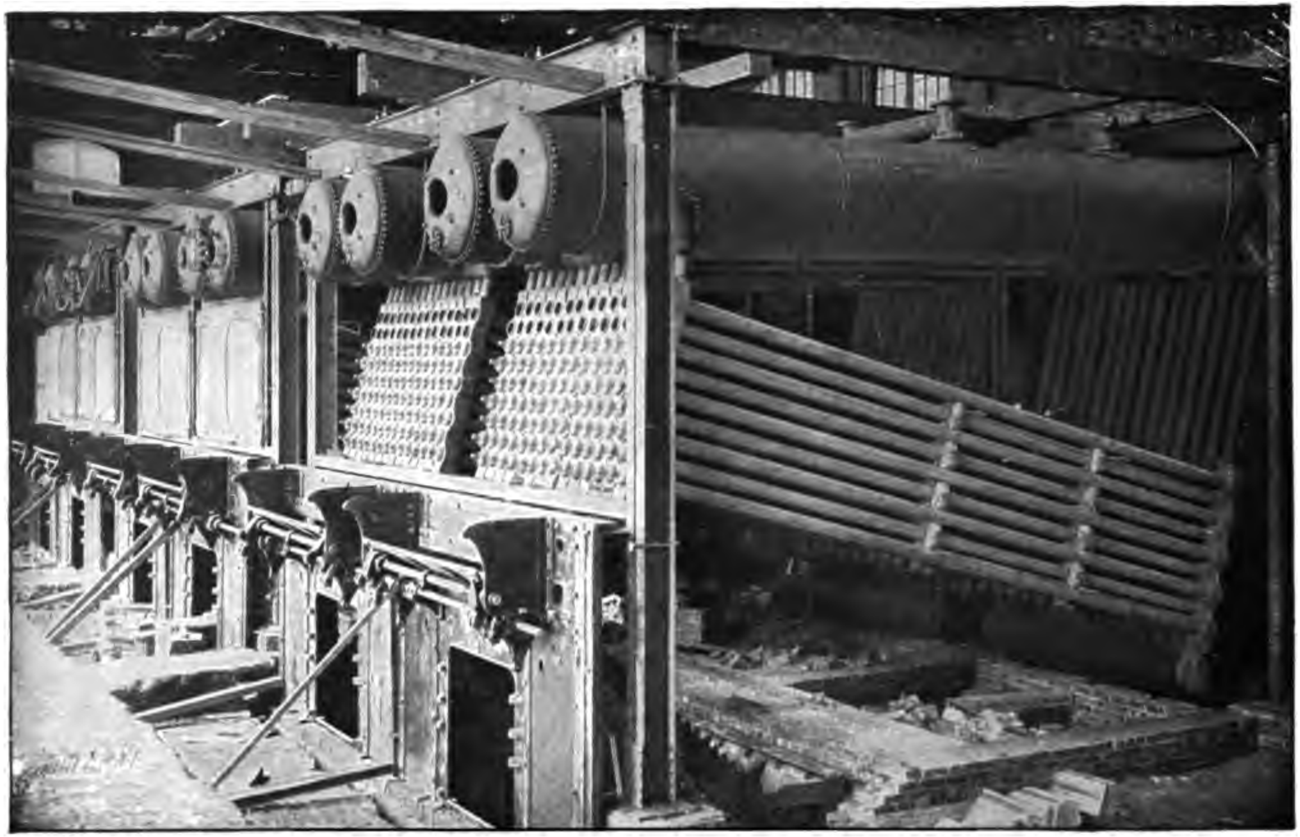


Fig. 2.



1600 Horse Power Babcock and Wilcox Boilers in Process of Erection.

The heating surface of this type is small with a given diameter unless the boiler is made very long. As all sediment collects in the bottom, where the heat is most intense, the plates are liable to burn. Since sediment and scale are poor conductors of heat, the heat remains in the plates and overheats them instead of flowing to the water.

The disadvantages (the small heating surface and the collection of sediment) do not seem so serious when one considers the simplicity of construction, strength, durability, and ease of repairing and cleaning.

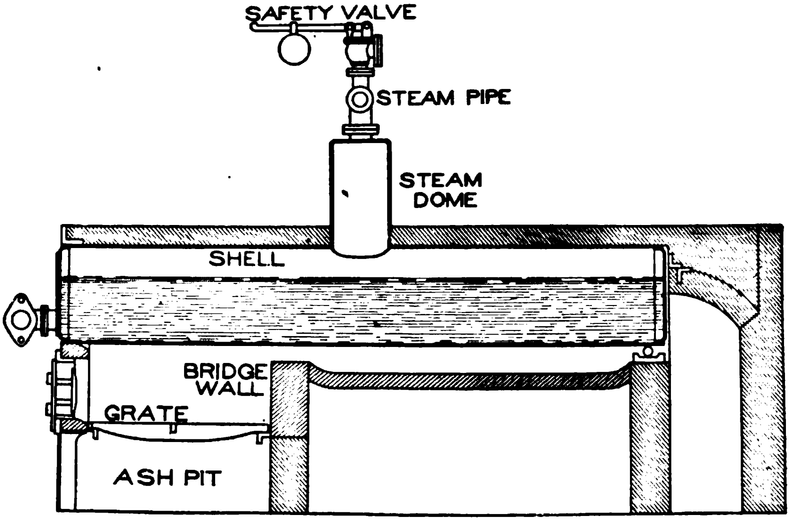


Fig. 3.

The plain cylindrical boiler was adapted for mining districts, iron works and other places where fuel is abundant and skilled boiler makers are not readily found. This boiler was made very long to get the required heating surface, the length sometimes exceeding fifty feet.

FLUE BOILERS.

In order to get the necessary heating surface in the cylindrical boiler without making it excessively long, it was made with an internal flue through which the hot gases passed to the chimney. This flue was quite large and extended from end to end. In the

United States, Oliver Evans used this type in 1800. In England, it led to the internally-fired flue boilers which were so extensively used.

THE CORNISH BOILER.

Horizontal—Single-Flue—Internally-Fired.

When it was found that about 25 per cent of the total heat of combustion was lost by radiation from the furnace, a Cornish engineer named Trevithick, conceived the idea of placing the fire inside the large internal flue. He introduced this type which is known as the Cornish boiler.

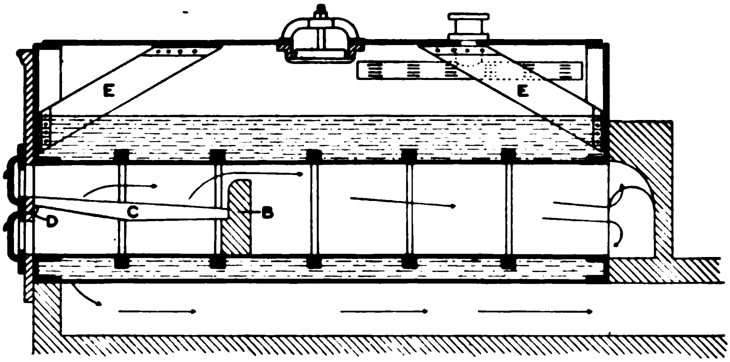


Fig. 4.

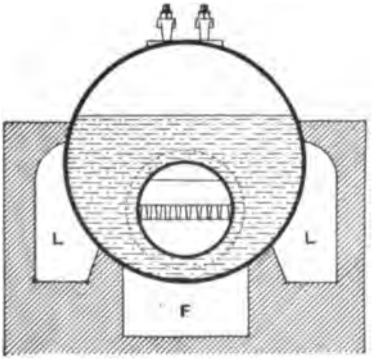


Fig. 4*a*.

The products of combustion pass from the fire on the grate bars C (Fig. 4) through the flue to the back end where they divide and return to the front end by means of the lateral flues L in the brickwork. See Fig. 4*a*. At the front the hot gases pass downward, and uniting pass through the flue F in contact with the bottom of the boiler. On leaving the boiler they go to the chimney. This arrangement of flues reduces the temperature of the gases before they come in contact with the bottom of the boiler where sediment collects. The grate bars rest on the dead plate D at one end and on the bridge B at the other; if made in two lengths (as is often the case) they are supported at the center by a cross bearer. The bridge is built of fire brick and the external flues are lined with fire brick. The heads are stayed to the shell by gusset stays E E.

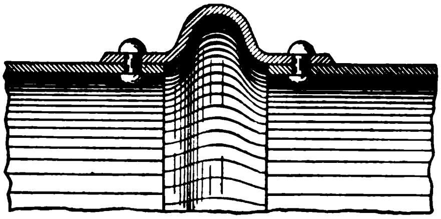


Fig 5.

The large internal flue is the hottest portion of the boiler because it contains the fire. For this reason the flue has greater linear expansion than the shell and, if the flue is a plain cylinder, the increase in length causes the ends to bulge. When the boiler is cold, the flue returns to its normal length. This lengthening and shortening will soon loosen the flue at the ends. To overcome this, the flue is sometimes made up of several short rings flanged at the ends and joined by being riveted to a plain ring. This construction is shown in section in Fig. 4. Another method is shown in section in Fig. 5. The plain ring is riveted to the curved ring; this ring takes up the expansion, increases the heating surface, and strengthens the flue against external pressure. The same results may be obtained by the use of the corrugated flue, one form of which is shown in Fig. 6. The corrugated flue has many advantages over the devices shown in Figs. 4 and 5; it is frequently used in marine boilers.

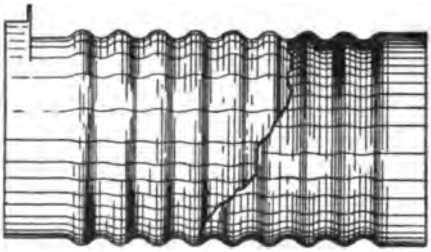


Fig. 6.

LANCASHIRE BOILER.

Horizontal—Two-Flue—Internally-Fired.

It can be proved, both by experiment and calculation, that with a given thickness large cylinders cannot stand as much external pressure as small ones. For this reason and on account of the short distance a fireman can throw coal accurately, the Cornish boiler is suitable for small powers only. If it is made too large, the flue is liable to collapse, but if, on the other hand, the flue is of too small a diameter, the grate will be insufficient. If this form of boiler is to be used in large size it is modified by using two flues instead of one. This boiler is called the Lancashire boiler. It is like the Cornish type except that it has two flues and, of course, two furnaces.

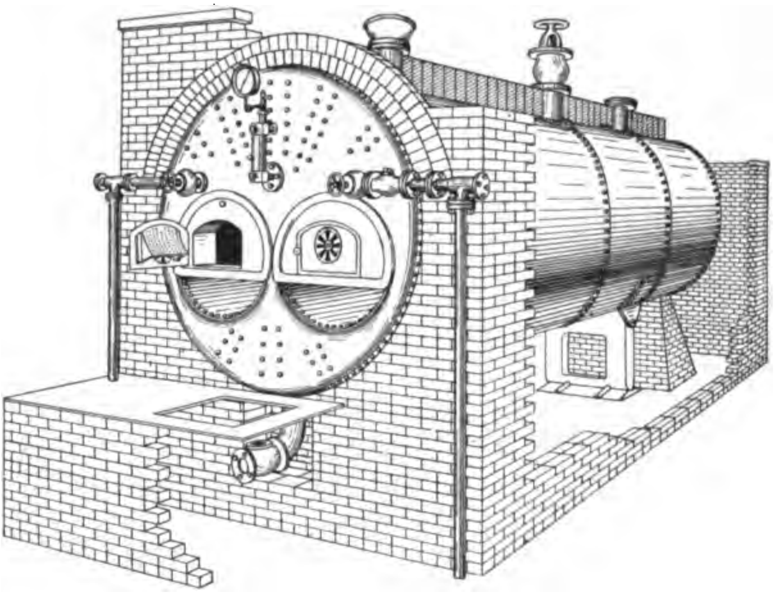


Fig. 7.

The flues are sometimes continued separately to the end. If they merge into one large flue, which forms the combustion chamber, it is called the "Breeches-flued" or duplex furnace boiler. These furnaces are fired alternately; the unburned gases set free from the freshly-fired coal are burned on meeting the hot gases from the incandescent coal of the other furnace. This arrangement prevents the escape of the unburned hydrocarbons.

The disadvantage of the Lancashire boiler is the difficulty in finding room for the two flues without greatly increasing the diameter of the boiler. Also, the small furnace is unfavorable to complete combustion as the space for the uniting and burning of the hydrocarbons is restricted. The combustion chamber of the breeches-flued boiler provides the necessary space, but the construction at the junction of the two flues is weak and has been the cause of many explosions.

GALLOWAY BOILER.

Horizontal—Two-Flue—Internally-Fired—Galloway Tubes.

Another boiler of the same general form is the Galloway, shown in Fig. 7. This boiler differs from the Lancashire in that short tubes are added to the flues. In the Galloway boiler having two distinct flues, the tubes were placed as shown in Fig. 8.

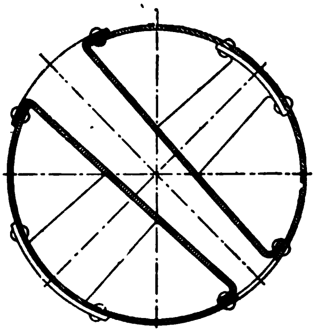


Fig. 8.

In the later form of Galloway boiler, the two flues merge into one large flue of the shape shown in Fig. 9. This flue has corrugated sides and the conical tubes are staggered, thus insuring a thorough breaking up of the currents of hot gases. The tubes are made conical to facilitate removal for repairs. The shape of the tube also permits the water to expand on being heated, and the par-

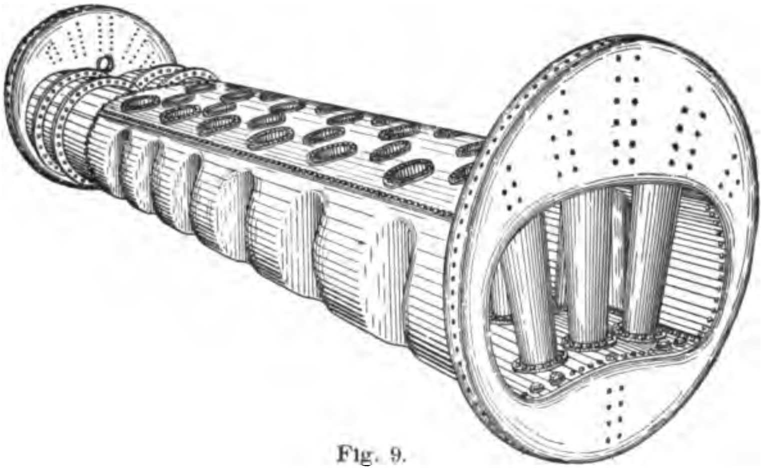


Fig. 9.

ticles rise vertically without disturbing the water on the heating surfaces above. The conical tubes are generally riveted rather than welded because the removal of a tube that is welded leaves a large hole in the flue.

FIRE-TUBE BOILERS.

SINGLE-FLUE BOILER.

Horizontal—Single Fire Tube—Externally-Fired.

In the Cornish, Lancashire, and Galloway boiler the large internal flue served as a fire box. There was, however, a flue boiler having the fire external to the shell. The boiler shown in Fig. 10 resembles the plain cylindrical boiler both in appearance and setting, but it has one or more large flues extending from end to end. This flue increases the heating surface to such an extent that the boiler can be considerably shorter than the plain cylindrical.

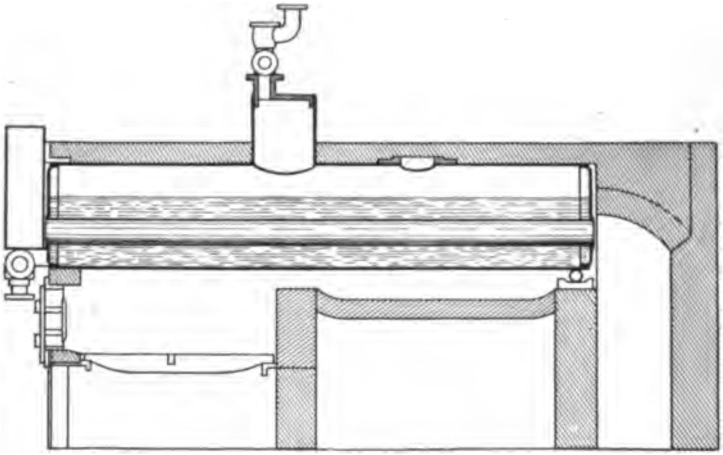


Fig. 10.

MULTITUBULAR BOILER.

Horizontal—Many Small Fire Tubes—Externally-Fired.

When engineers found that the internal flue was such an advantage (that is, it increased the heating surface), they soon added more tubes; as the number increased, the size diminished until they became of the size used at present. This is in brief the development of the multitubular boiler. This type of boiler has for many years been commonly used for stationary work and although other types possess advantages for certain conditions, it is still considered economical, reliable, easily handled, and safe if constructed of good material and operated with care and intelligence.

Figs. 11 to 14 are selected to illustrate this boiler. The boiler without the brick setting is shown in Fig. 11. It consists of a steel cylindrical shell and numerous small tubes extending from end to end. These tubes are 3 or 4 inches in diameter and are fastened to the two ends (called tube sheets) by expanding the tubes against the sheet and beading them over on the outside. The shell is made of steel plates $\frac{1}{4}$ to $\frac{3}{4}$-inch in thickness. At the front, the shell plates extend beyond the tube sheet and are cut away to allow the waste gases to enter the uptake. About one-third the volume of the boiler is occupied by the steam; the other two-thirds is filled with water and tubes. The water line is a little (from 4 to 8 inches) above the top row of tubes.

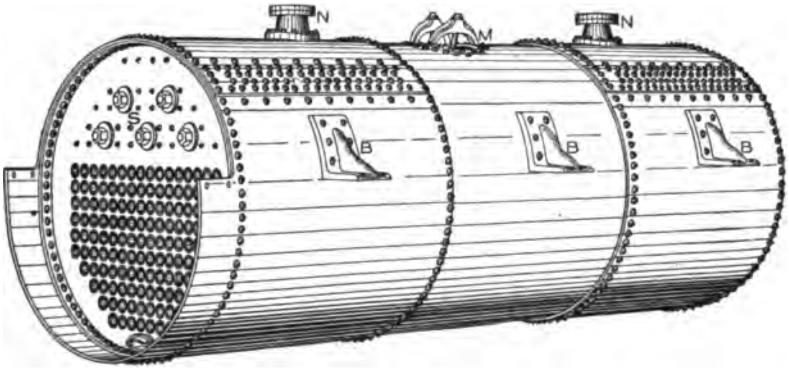


Fig. 11.

The flat ends are prevented from bulging by stays which may be of the form shown in Fig. 12 or they may be diagonal stays. The through stays are fastened to the tube plates by means of nuts and washers as shown at S in Fig. 11, and also in Fig. 12. Below the water level, the end plate is stayed by the tubes. This type of boiler may be supported by brackets B riveted to the shell or by means of beams and columns, as shown in Fig. 14. The front bracket is often fixed in the side wall, but the rear bracket should be placed on rollers so that it can move on an iron plate. This

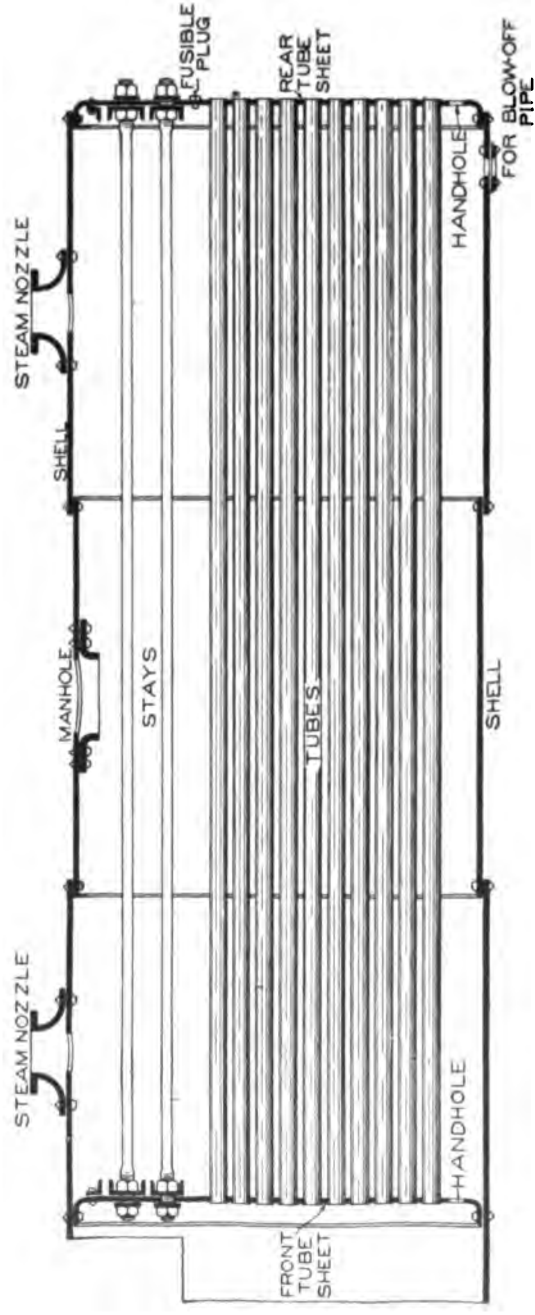

FUSIBLE PLUG
REAR TUBE SHEET
FOR BLOW-OFF PIPE
HANDHOLE
STEAM NOZZLE
SHELL
MANHOLE
STAYS
TUBES
SHELL
STEAM NOZZLE
HANDHOLE
FRONT TUBE SHEET

Fig. 12.

will prevent the straining of the plates from expansion and contraction. A small space must be left between the rear tube sheet and the brick wall to allow for expansion.

The boiler shown in Fig. 11 has two steam nozzles N. If the boiler has a dome (D Fig. 13) the steam nozzle is at or near the top of the dome The feed pipe may enter either at the front or at the rear. It frequently terminates in a perforated pipe below the water line. The blow-off pipe is at the rear of the boiler as shown in Fig. 13. A valve, called the blow-off valve, regulates the flow and may be opened, when there is low pressure in the boiler, to blow out sediment and detached scale. The boiler is usually set with a slight inclination toward the rear so that mud and detached scale may collect near the blow-off pipe.

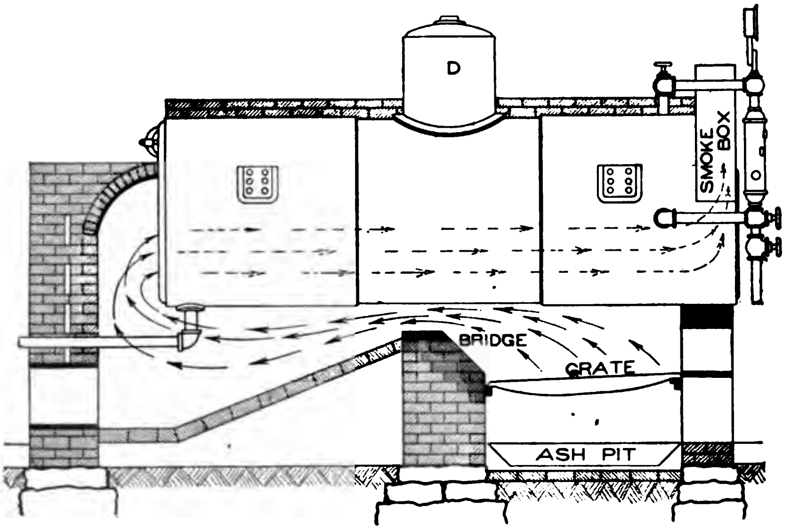


Fig. 13.

In order that the boiler may be entered for cleaning or repairs, it is provided with manholes and handholes. Fig. 11 shows a manhole M at the top near the middle and a handhole near the bottom of the front tube sheet. Handholes may be put in wherever desired, but manholes can be located only where the arrangement of stays and tubes will permit the entrance of a man. Manholes and handholes are elliptical; the former being about 11 inches by 15 inches in size; while the latter are about 4 inches by 6 inches.

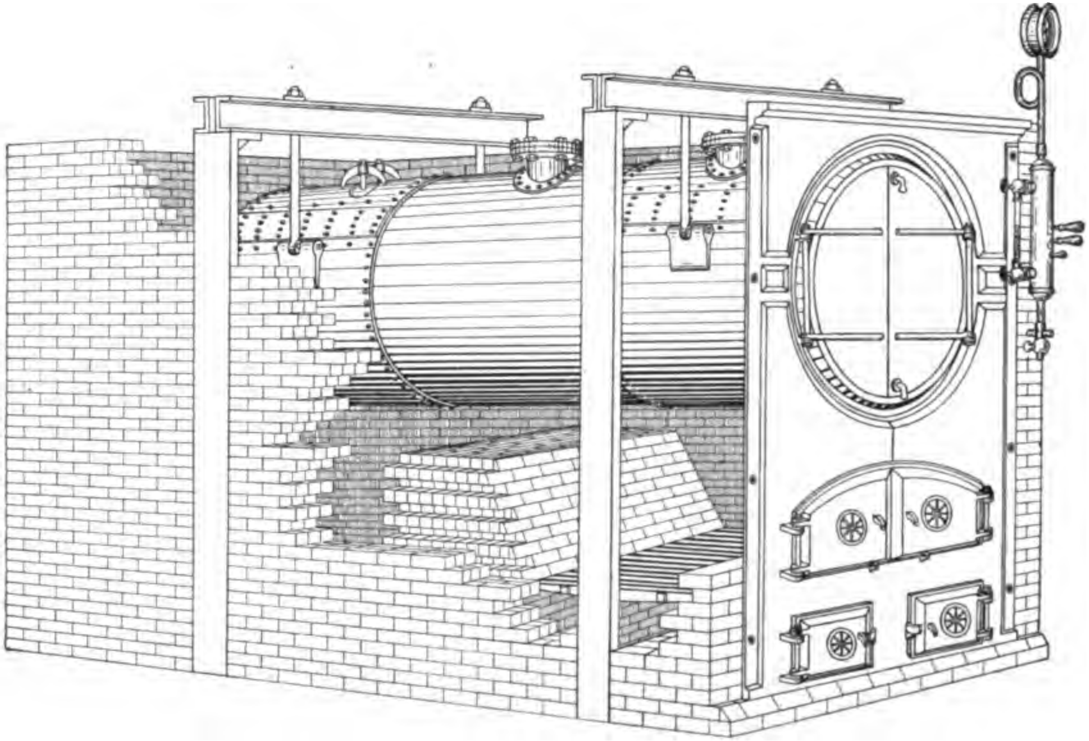


Fig. 14.

The heating surface is the surface in contact with the hot gases. In this type, the heating surface is made up of about half the shell, the tubes, and about two-thirds of the rear tube sheet. In general, all the heating surface is below the water line.

The complete multitubular boiler is shown in its brick setting in Fig. 14, and a longitudinal section of the setting in Fig. 13. The brick setting consists of brick laid in cement or mortar. The bridge and the portions of the furnace exposed to the fire are lined with fire brick. The bridge is built at the rear of the grate and forms a support for the grate bars; it also directs the flames upward. The arrows show the direction of the flow of hot gases. The furnace is formed by the bridge, the side walls, and the lower part of the boiler front. The boiler front is usually of cast iron with the lower part lined with fire brick. The front has doors which lead to the furnace, ashpit, and smoke box. The space below the grate is called the ashpit, and through its doors ashes are removed and a large portion of the air for combustion enters. Both the fire doors and ashpit doors have draft plates, or grids, to regulate the supply of air. The doors to the smoke box give access to the tubes for cleaning and repairs.

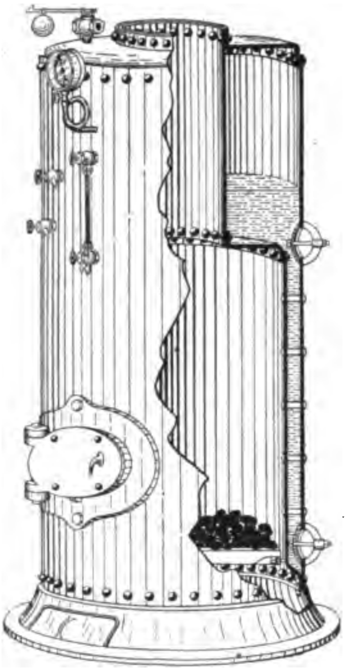


Fig. 15.

UPRIGHT BOILERS.

Vertical—Many Small Fire Tubes—Fire Box.

Upright boilers are used when floor space is valuable and there is sufficient height. In small sizes, they are used for hoisting engines, pile driving, for supplying steam for pumps, and similar work; in large sizes when it is necessary to have a powerful battery in a small space. In general they are not as economical as the horizontal multitubular boiler unless they are carefully designed and of considerable height. If the tubes are short, the hot gases escape before they give up much of their heat.

One of the simplest forms of upright boiler is shown in Fig. 15. It has a cylindrical shell with a large fire box at its lower end. This fire box is formed by the inner cylinder which is fastened to the outer shell by short screw stay bolts as shown. A flanged ring connects the fire box with a large flue which conducts the hot gases away. The necessary handholes, gages, safety valves, etc., are provided. This form is not economical but is used on account of the little attention required.

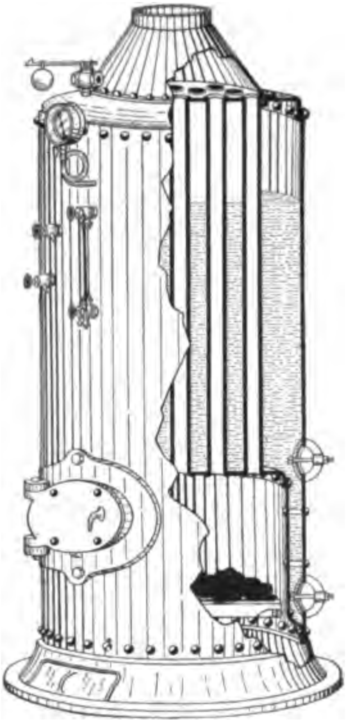


Fig. 16.

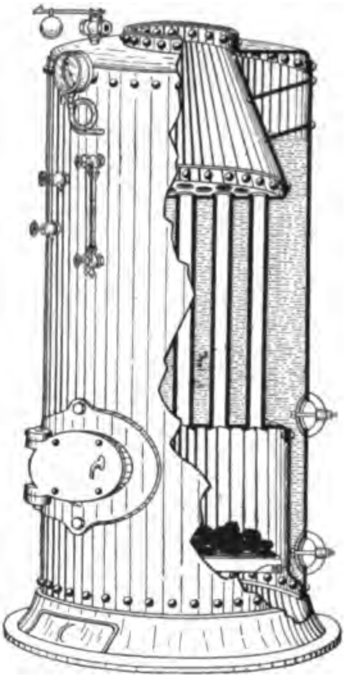


Fig. 17.

More economical forms of the small upright boiler are illustrated in Fig. 16 and 17. The boiler shown in Fig. 16 is a common form; externally it is like the boiler represented by Fig. 15, but within, it has a somewhat different construction. It resembles a multitubular boiler placed on end. The fire box is made of an inner cylinder stayed to the outer. The top of the fire box, called the lower tube sheet, is connected to the upper head by tubes, through which the hot gases pass to the smoke pipe. It will be readily seen from Fig. 16, that the upper ends of the tubes are surrounded with steam while the lower portions are covered with water. As steam is a poor conductor of heat, the ends of these tubes are liable to injury from overheating.

In the class of boiler shown in Fig. 17 the upper ends of the tubes are below the water level, thus avoiding the weakness described in connection with Fig. 16. The upper tube sheet is submerged and is flanged and riveted to the frustum of the cone which forms the smoke box. The chief defect in this boiler is that the lower part of the cone is often placed too near the shell; this is done to admit more tubes. This construction restricts the space so much that there is not sufficient room for the steam to rise as it is formed on the tubes. The cone, which is subjected to external steam pressure, is likely to be weak and is usually carefully stayed.

These small upright boilers require no brick setting, as the fire box is within the boiler and the cast-iron foundation forms the ashpit.

MANNING BOILER.,

Vertical—Many Small Fire Tubes—Fire Box.

The Manning boiler is illustrated in Fig. 18. In order to get a large heating surface, it is made 20 to 30 feet high. It is, in general, similar to the upright boiler shown in Fig. 16. At the lower portion, the shell is of greater diameter than at the top in order to provide a large grate area. The inner fire box is stayed to the shell by screw stay bolts. As the fire box is surrounded by water and there are many long tubes there is a large heating surface. The tubes are arranged in concentric circles with a space for circulation in the middle.

The external fire box is joined to the shell by a double flanged ring as shown in Fig. 19; or, by the cone-shaped section as illustrated in Fig. 20. The top edge of the internal fire box is riveted to the lower tube sheet which is flanged. The bottom of the inner fire box is connected to the outer shell by a welded ring (shown

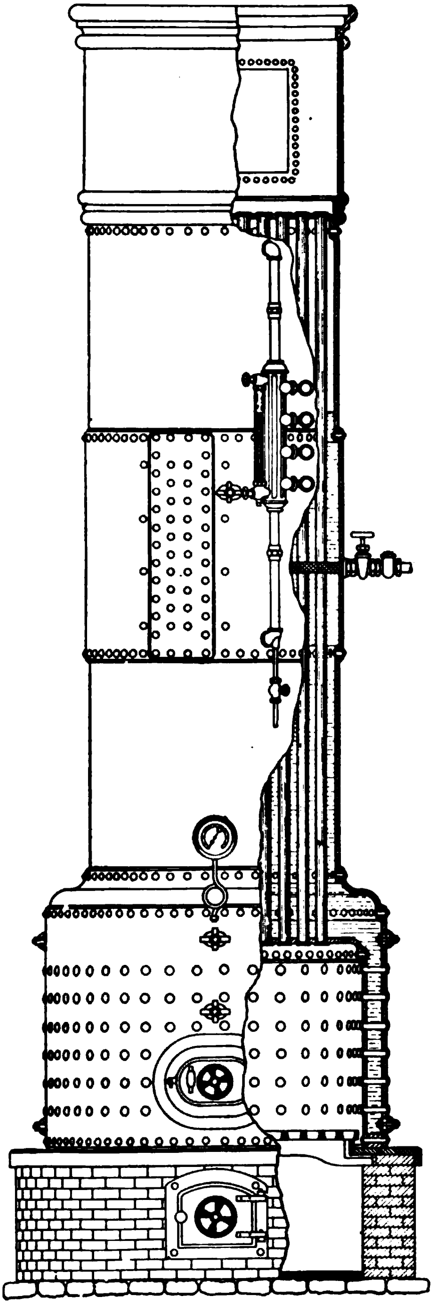


Fig. 18.

in section in Figs. 19 and 20) called the foundation ring. The water space between the inner and outer fire box plates, called the water leg, should be large.

This boiler is cleaned by means of handholes. They are placed in the shell plates near the lower tube sheet, in the external fire box just over the furnace door, and at the bottom near the foundation ring. As there are no manholes for cleaning, the boiler is suited to good feed water only.

The feed pipe enters the shell at the side near the middle of the water space, and extends across the boiler; it is perforated to distribute the water.

The heating surface consists of the inside of the fire box and the tubes up to the water level, and the tube sheet. That part of the tubes *above* the water line is the superheating surface; that is, the heat from the gases passes through the metal of the tubes to the steam, thus raising its temperature without raising its pressure. Steam heated under these conditions is called superheated steam. In small vertical boilers this superheating surface is not desirable because the work of the small boiler does not require superheated steam and the tubes are likely to be burned by the intense heat. With the long tubes of the Manning, the gases are not as hot when they reach the top, and as this boiler is built in large powers (200 horse-power being common) the engines supplied are built for economy and require dry if not superheated steam.

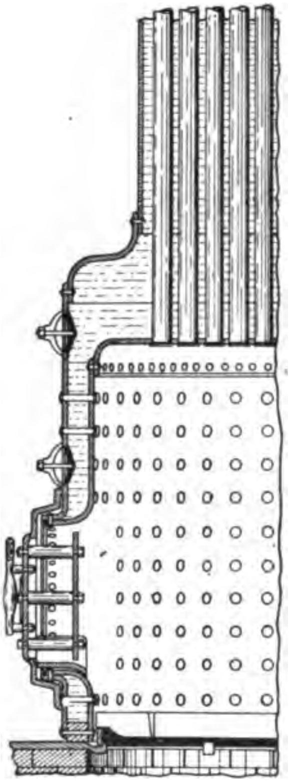


Fig. 19.

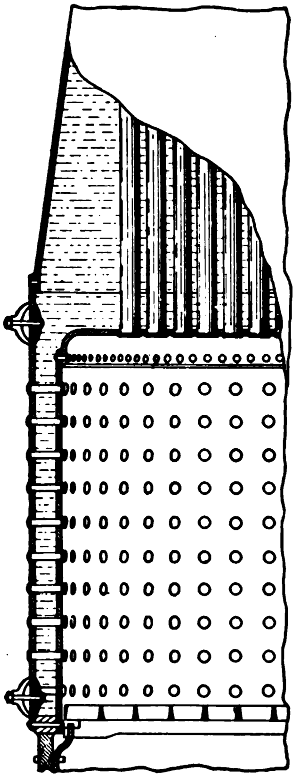


Fig. 20.

RETURN-TUBE BOILERS.

Horizontal—Many Small Fire Tubes—Internally-Fired.

The boilers hitherto described are used mainly for stationary work, the exceptions being so few that they need not be even mentioned. Let us now discuss another modification of the fire-tube boiler—one that has been and is now extensively used in marine work. The parts of the return-tube boiler are essentially the same as those of flue boilers (Cornish and Lancashire) and the multitubular boiler. They are, however, arranged differently in order to be used on board ship.

The earliest forms of marine boilers, working with pressures of 15 to 30 pounds per square inch, were square or box-shaped. They were economical and of convenient form for ships. When higher steam pressures became necessary, the flat surfaces required so much staying that they were abandoned and the cylindrical type introduced, as this form is the best of the practical shapes to resist internal pressure. The cylindrical form may not be as conveniently stowed aboard ship, but it will stand much higher pressures. The cylindrical marine boiler is frequently built for 170 pounds per square inch.

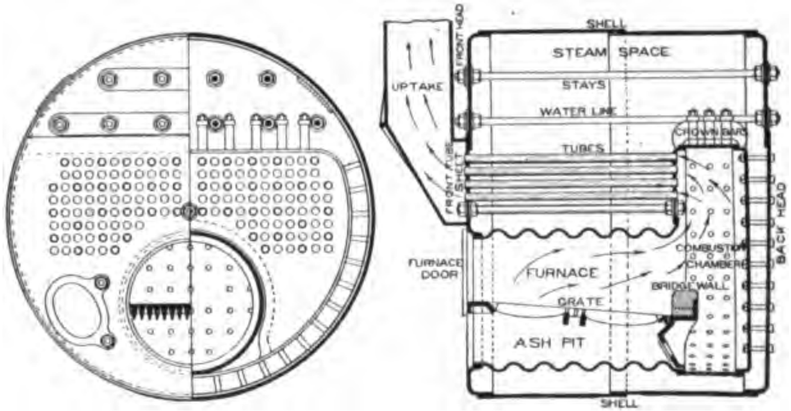
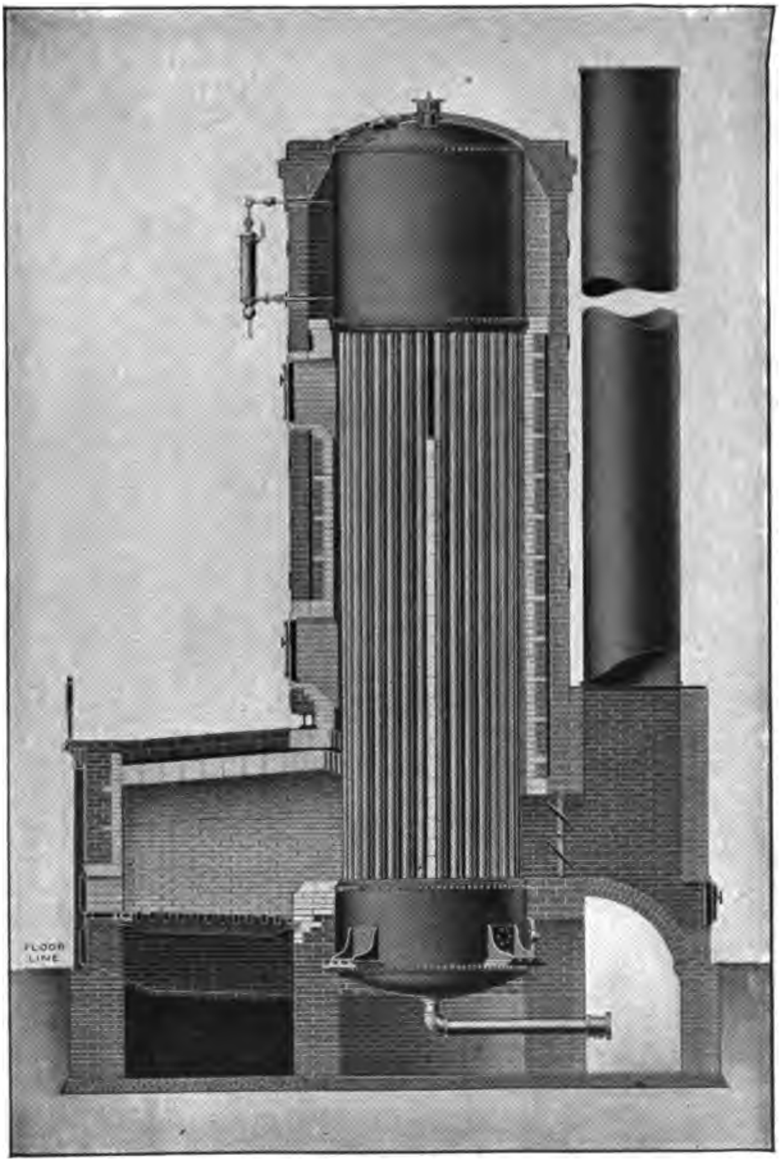


Fig. 21.

The single-ended, return-tube boiler, shown in Fig. 21, combines the internal furnace flue of the Cornish type and the numerous small fire tubes of the multitubular. The cylindrical shell is made up of plates riveted together and to the flat ends of the boiler, which are flanged to fit the shell.

The furnace is cylindrical, three to four feet in diameter and about seven feet in length. The front end of the furnace flue is riveted to the front end plate, which is flanged for the purpose. The back end is riveted to the combustion chamber plates. For-



THE WICKES VERTICAL WATER TUBE BOILER.

merly, the flue was a plain cylinder, but as a plain cylinder, unless of small diameter, cannot stand much external pressure, it soon became necessary to strengthen it. This was done by means of the curved ring shown in Fig. 5 and other methods; but at present the corrugated flue is used, one form being shown in Fig. 6.

The grate is placed at about the center of the height of the furnace flue; the space above this grate is occupied by the fire and hot gases, below it is the ashpit. As will be seen from the arrows in Fig 21, the hot gases fill the space above the fire, the combustion chamber, the tubes and the uptake.

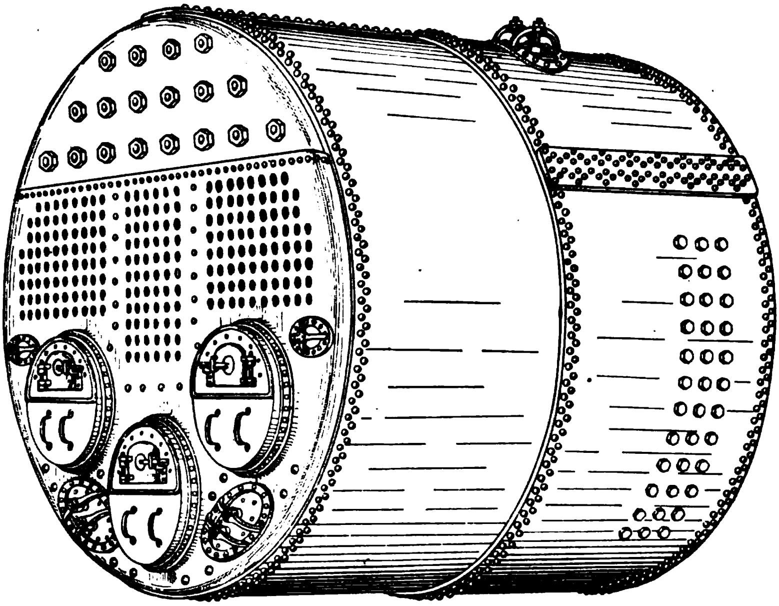


Fig. 22.

The combustion chamber in which the products of combustion are completely burned, is formed of flat and curved plates flanged at the edges and riveted together. The shape of the plates is shown in Fig. 21, which is a sectional view of a single-ended marine boiler. The back tube sheet forms the front of the combustion chamber. The space around the tubes, furnace flue, and combustion chamber is filled with water, the water level being six to eight inches above the top row of tubes. The space above the water level is called the steam space.

As the return-tube boiler has several flat surfaces, this type requires careful staying. The flat ends above the water level are prevented from bulging by long stay rods which are similar to those in the multitubular type. Below the water level, the furnace flue and the tubes aid in holding the flat plates together. In addition, a few of the tubes (shown by the heavier circles in Fig. 21) are made thicker so that a thread may be cut on the ends which are screwed into the tube sheets and held by thin nuts. The combustion chamber plates are stayed to the rear end plate and the shell by short screw stay bolts. The flat top of the combustion chamber is supported by girders or crown bars.

Number of Furnaces. The boiler shown in Fig. 21 has only one furnace, but return-tube boilers frequently have two, three, or four furnaces.

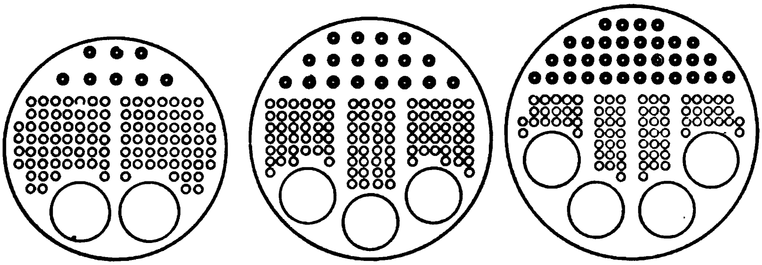


Fig. 23.

Fig. 22 shows a boiler with three furnaces. Large furnaces are more efficient than small ones because the grate area increases directly as the diameter, while the air space above the grate increases as the square of the diameter. The greater space aids combustion. The length of the grate bars is nearly constant for all sizes of flue because it is limited by the distance a fireman can throw coal. Furnace flues are usually from 36 to 54 inches in diameter. As the size of furnaces is fixed, the number depends upon the size of the boiler, for a large boiler must have a large grate area which can be obtained only by using several furnaces. The various arrangements are shown diagrammatically in Fig. 23.

A single-furnace boiler has but one combustion chamber. A two-furnace boiler may have a combustion chamber for each furnace or it may have a common combustion chamber. If there is but one boiler on board, it is better to have two combustion chambers, so that in case a tube bursts, the boiler will not be disabled. If, however, there are several boilers, it is better to have a common combustion chamber for the two furnaces, because the alternate stoking keeps up a more nearly constant pressure of steam and there is less smoke. Three-furnace boilers usually have three combustion chambers, while four-furnace boilers have two. In case four furnaces are used with three combustion chambers, the two center furnaces lead to a common combustion chamber and each outside furnace has one.

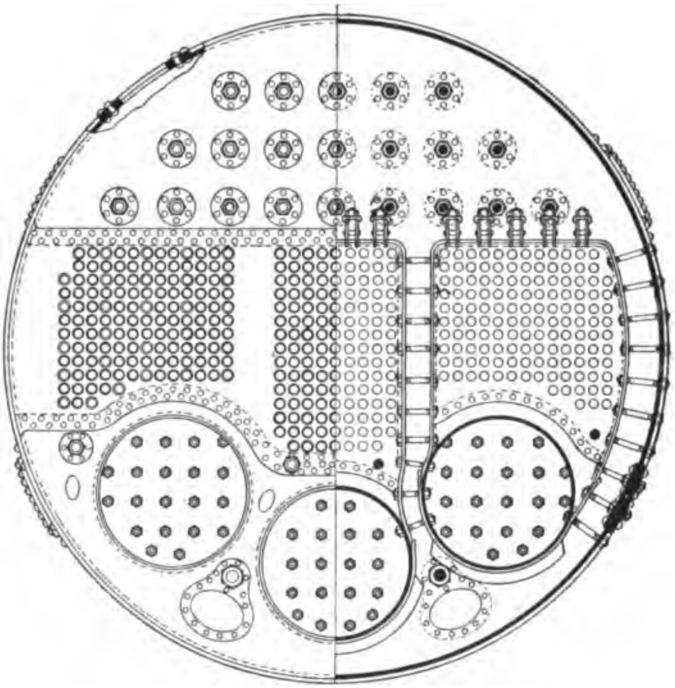


Fig. 24.

Double-ended Boilers. This form of marine return-tube boiler is practically the same as two single-ended boilers placed back to back, but with the rear plates removed. The weight of the rear plates is saved and there is less loss from radiation. This makes the double-ended boiler lighter and cheaper in proportion to the heating surface. Double-ended boilers are often made 16 feet in diameter and 18 feet long.

There are two distinct classes of double-ended return-tube boiler—those having all the furnaces open into one combustion chamber and those having several combustion chambers. The boiler having but one combustion chamber has the disadvantage that if one fire is being cleaned the whole boiler may be cooled by the inrush of cold air. It is better to have a combustion chamber for each furnace or at least have a combustion chamber for the furnaces of each end. The usual method of dividing up the combustion chambers is by water spaces as shown in Fig. 24, which is the section of a boiler having a combustion chamber for each furnace.

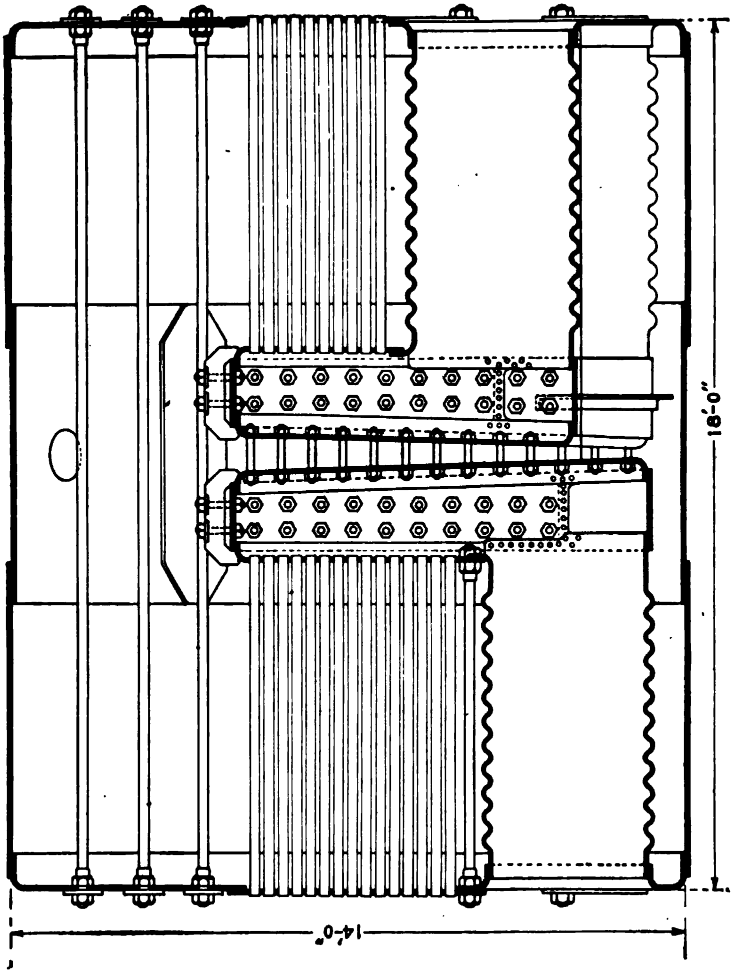


Fig. 24.

Internal Furnace Return-Tube Boiler. Although the return-tube boiler is commonly used in marine work, this type, with some changes in detail, is used in plants ashore. Fig. 25 shows the construction and arrangement of parts. The flue is larger in proportion to the diameter than is the case with the marine form; the combustion chamber is partly external to the shell, that is, the rear tube sheet is also the rear end plate. This arrangement does away with the necessity of staying the flat plates of the combustion chamber.

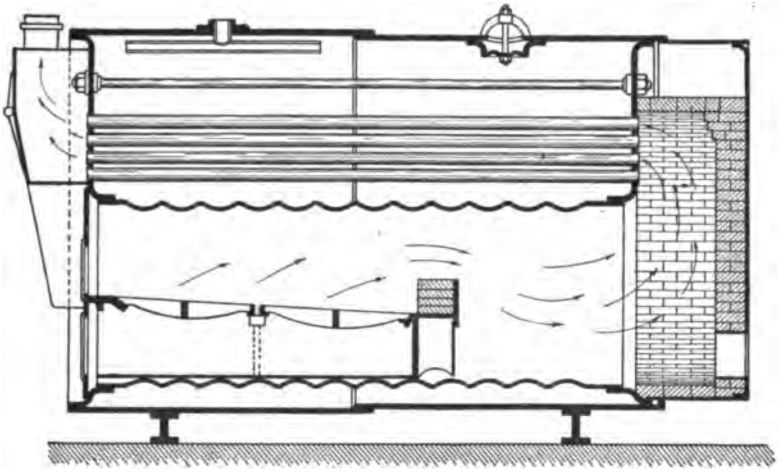


Fig. 25.

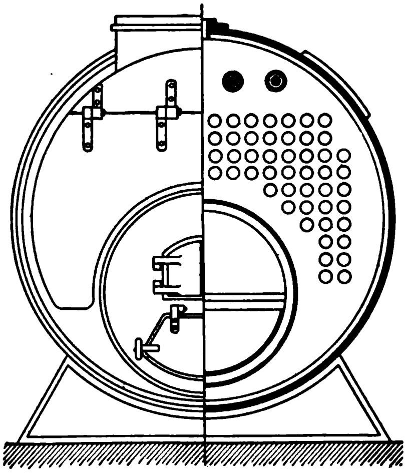


Fig. 25.

Another form of internal furnace, return-tube boiler is shown in Fig. 26. This boiler usually has two flues extending from the front to the back head. The grate is placed in the corrugated portion while conical water tubes support the flue back of a bridge wall. The large furnaces and the space around the conical tubes provide a combustion chamber of ample size.

The arrows show the direction of the hot gases. After leaving the internal flue they enter the return tubes which are below the furnace; before leaving the boiler, they pass underneath the shell. By this arrangement the hottest gases are near the water line and the cooler gases in contact with the cold water, thus there is the greatest difference in temperature at all times. At each change in the direction of the hot gases, there is an opportunity for dirt and ash to fall by gravity so that the tubes may remain clean and efficient.

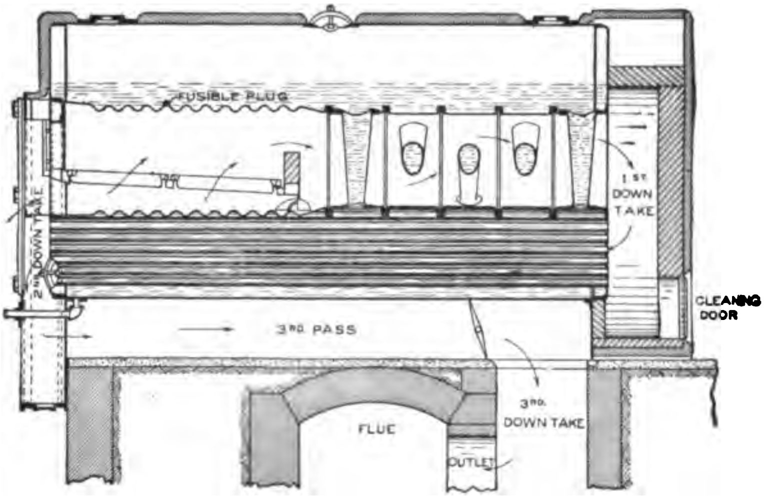


Fig. 26.

With the exception of the foundation there is no brickwork. The shell is covered with a non-conducting material. This boiler, like the Galloway, has a large steam and water space, thus insuring dry steam and great reserve power.

THROUGH-TUBE BOILERS.

Horizontal—Many Small Fire Tubes—Internally-Fired.

Vessels of slight draft require a boiler of small diameter. This is especially true of gunboats as it is desirable to have the boilers below the water line. As there is not room for the return-tube boiler, the through-tube, shown in Fig. 27, is sometimes used. This boiler is made up of the same parts as the return tube, the chief difference being that of arrangement. The rear plate of the combustion chamber forms one tube sheet and the end plate forms the other. The top of the combustion chamber is stayed to the shell by sling stays which are bars having forked ends fastened to the shell and to the combustion chamber.

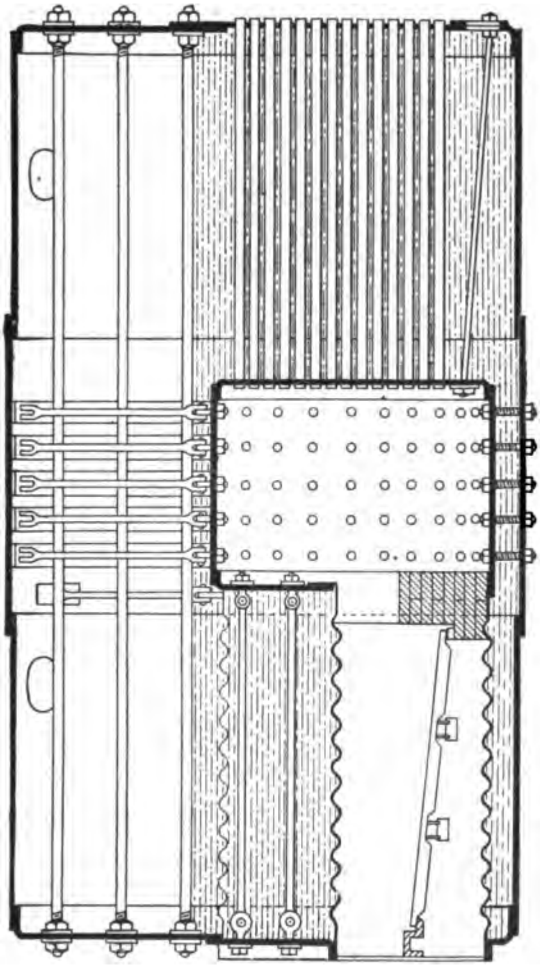


Fig. 27.

The fire is in a flue, or flues, which leads to the combustion chamber. The hot gases pass from the combustion chamber through the tubes to the uptake at the back end. The chief objection to this form is its length, for the heating surface is small unless the boiler is made very long.

FIRE-BOX BOILERS.

LOCOMOTIVE TYPE.

Horizontal—Many Small Fire Tubes—Externally-Fired.

Although vertical fire-tube boilers may be classed as fire-box boilers, yet the name fire-box boiler is usually applied to the locomotive type whether used with a locomotive or as a stationary boiler.

The usual form of horizontal fire-box boiler consists of a cylindrical shell, or barrel, partly filled with tubes, and a rectangular fire box. The shell is prolonged beyond the rear tube sheet to form a smoke box. The front ends of the tubes open into the fire-box, while the rear ends open into the smoke box. The hot gases from the fire pass through the tubes to the smoke box and from thence to the stack or uptake. For locomotive work, there are a large number of small tubes (usually 2-inch), but for stationary work the tubes are larger and less numerous. The reason for this difference is that in the locomotive boiler a greater heating surface is necessary, and to obtain sufficient draft to burn the large amount of coal for this heating surface, the exhaust steam is turned into the smoke box. The blast of steam carries the heated gases up the stack and a fresh supply of air passes through the grate.

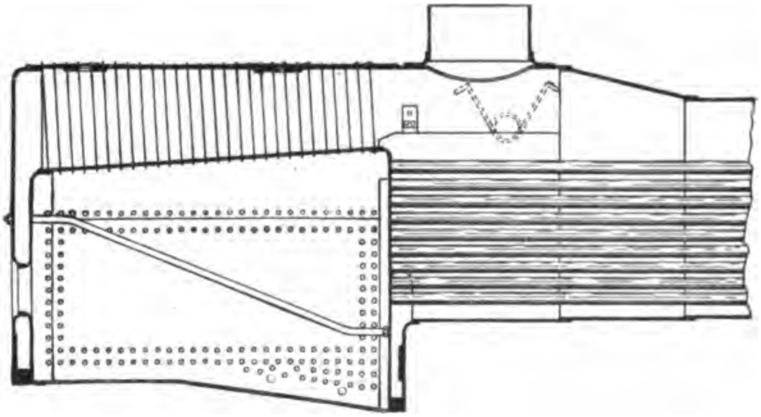


Fig. 28.

The cylindrical shell is joined to the fire box by riveting to a flanged ring or to a cone-shaped portion as in the vertical boiler. The fire box has a rectangular cross-section and usually a flat top.

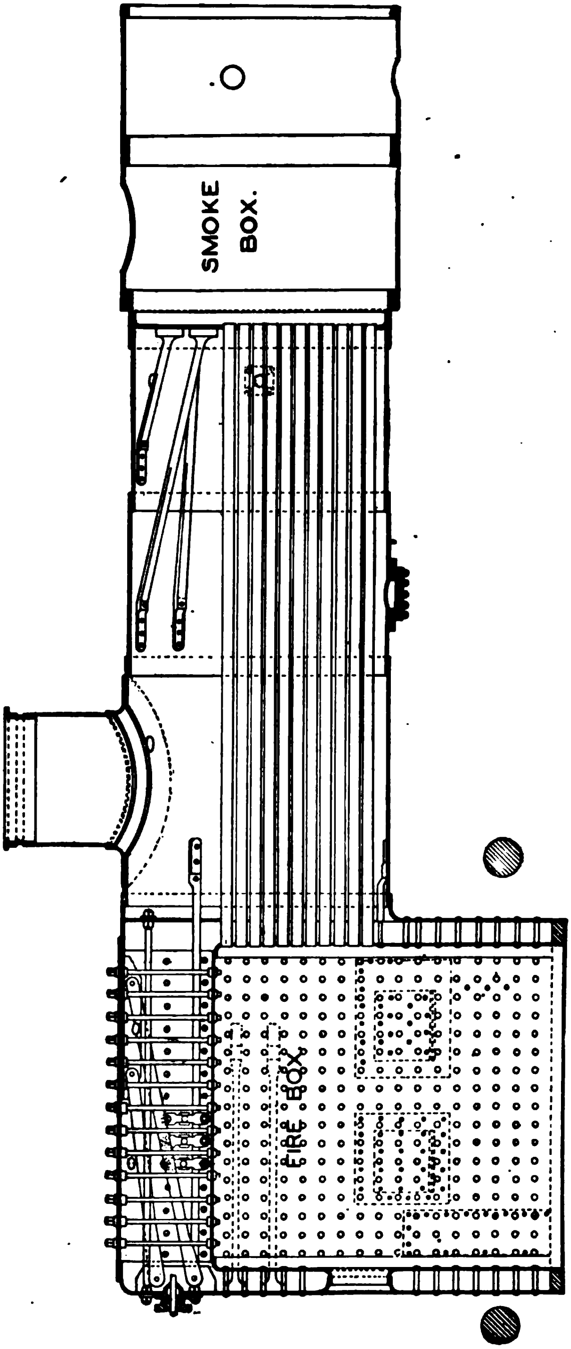


Fig. 29.

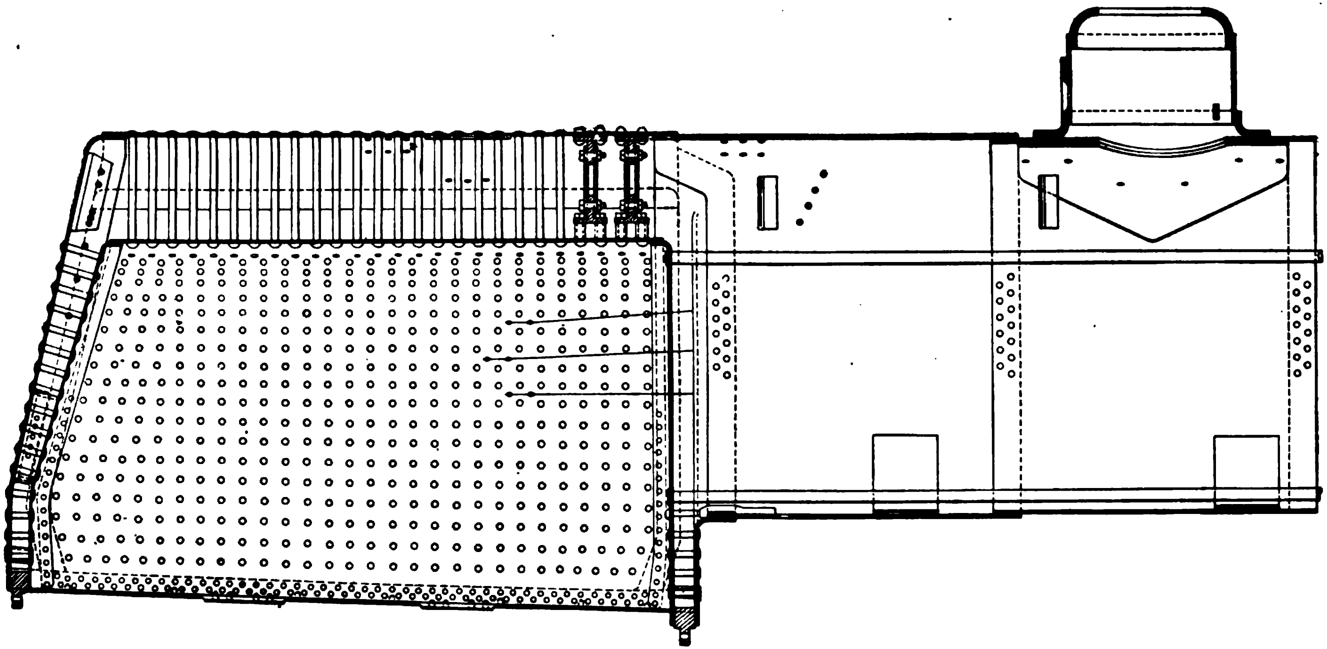


Fig. 30.

Like the vertical boiler there is an inner and an outer fire box, the inner having the same shape as the outer, except that the top is flat. The external fire box is connected to the inner by short screw stays. The space between is called the water leg. The flat top is stayed by girders or crown stays. These are sometimes attached to the shell by sling stays. The lower portions of the tube sheets are held in place by the tubes; the upper portions are stayed by diagonal stays.

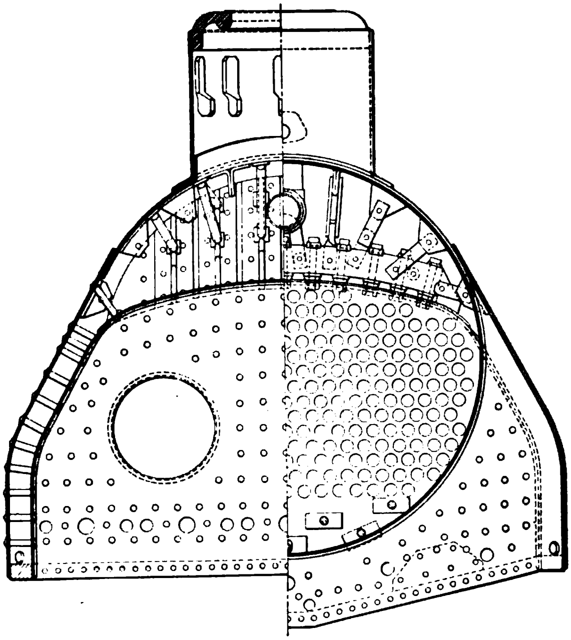


Fig. 31.

The chief differences in the various forms of locomotive boilers are the shape of the fire box and the location of the grate. Locomotive boilers are either straight top or wagon top. The wagon top boiler, see Fig. 28, has a cone-shaped portion by means of which the boiler is larger at the fire-box end. This construction is to give a greater steam space. The increase in size of boilers has raised the top so high above the rails that the wagon top is not now used extensively; the straight top, see Fig. 29, is more common.

Belpaire Boiler. The shell and fire tubes of this type of boiler are practically the same as in any other fire-box boiler; the peculiarity lies in the fire box. The inner and outer fire-box plates are horizontal at the top, and the sides of the outer fire-box are continued so that the space above the crown sheet is rectangular in section. The advantage of this construction is that the staybolts holding the crown sheets and side sheets can be placed at right angles to the sheets. This reduces the tendency to bending when under pressure.

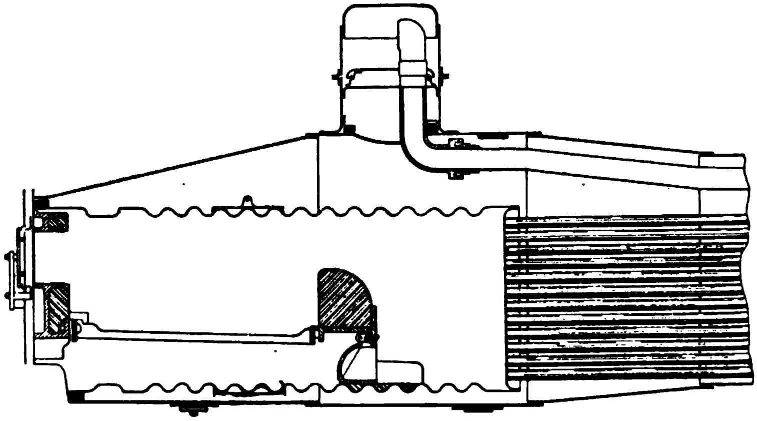


Fig. 32.

Wootten Boiler. In this type also, the fire box is the chief portion to be considered. The size of the locomotive fire box is limited. With older types the width was limited to less than three feet and the length to less than seven feet. This was because of the frames and the distance between the axles. By placing the fire box above the axles, the width was increased by an amount equal to the thickness of the frames or about seven inches, and the length increased to about eleven feet. By raising the fire box still more and placing it above the driving wheels, the width can be still further increased.

A broad, shallow fire box is required if anthracite coal is used. The Wootten fire box, shown in Fig. 30 and 31, is very wide and is placed on top of the driving wheels. Formerly, a combustion chamber was placed between the end of the grate and the tubes, but as it was found to be unnecessary, it is not now used.

Lentz Boiler. The object of the design shown in Fig. 32 is to avoid the use of stays. To do this no flat plates are used, except the tube sheets and these are stayed by the tubes. The firebox is in the form of a corrugated flue similar to those in internally-fired, return-tube boilers. As this is circular it requires no stays. The shell is circular and shaped as shown in the illustration. This type has been much used in Europe, but a few have been built in this country.

FIRE-BOX BOILER FOR STATIONARY WORK.

The fire-box boiler, usually called the locomotive boiler, is often used for stationary work, traction engines, and for vessels of light draft. This type of boiler, slightly modified, is sometimes used for generating the steam for heating buildings. It is economical and durable when used with natural draft. The chief differences in construction are larger tubes because of the draft, and the changes due to method of support. A common form is shown in Fig. 33. This type has been built in large sizes for high pressure, but when so made is expensive.

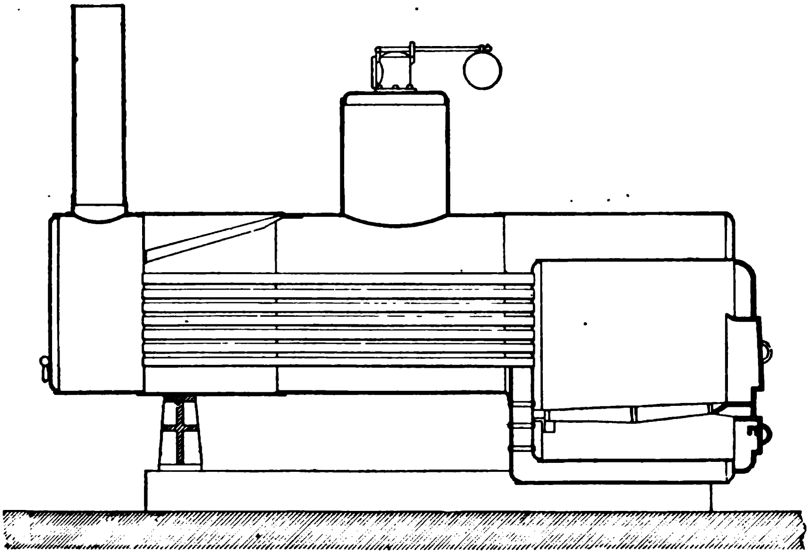


Fig. 33.

PECULIAR FORMS.

Fire-Tube Boilers, but differing from those described.

Return Tubular as Stationary. Boilers of the form shown in Fig. 34 resemble the locomotive, fire-box type, but in addition have return tubes. The hot gases reach the uptake by means of these tubes instead of passing to the chimney from the smoke-box end. Thus they combine the advantages of the fire-box type and the return-tube type without the brick furnace. The water surrounds the furnace on all sides except the front. They are built in sizes from 12 to 70 horse-power. As Fig. 34 shows the construction so clearly, further description is unnecessary.

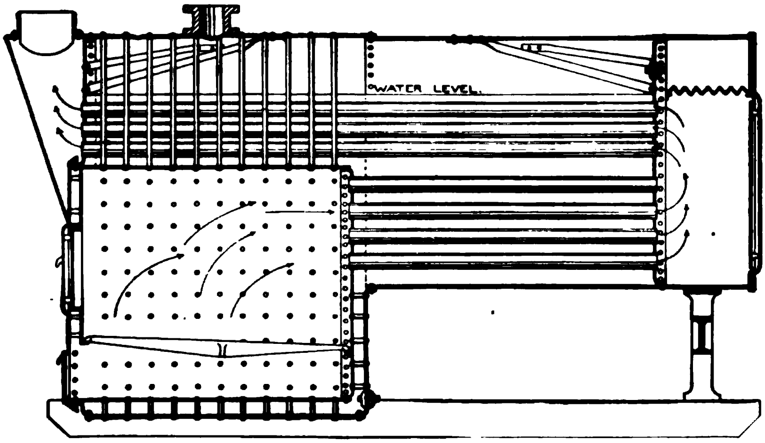


Fig. 34.

The Cochrane Vertical Boiler is somewhat like the return-tube boiler in point of arrangement of heating surface. This boiler is shown in section in Fig. 35. The hot gases pass from the furnace to the combustion chamber, then through the tubes to the uptake. The heating surface consists of tubes and the plates of the fire box which is surrounded by water except the bottom. The crown of the boiler and of the fire box, being hemispherical, require no staying. The hemispherical crown also allows a large steam space. The flat plates (the tube plates) are held together by the tubes.

The Shapley Boiler, shown in Fig. 36, may be called a return-flue vertical boiler. The upper portion is a reservoir for water and steam and the lower contains the fire box. The crown sheet of the fire box is stayed to the top by through stays. The hot gases from the fire rise in the fire box, pass through short horizontal tubes to an annular space. This annular space is connected to the flue at the base by vertical tubes passing through the water space.

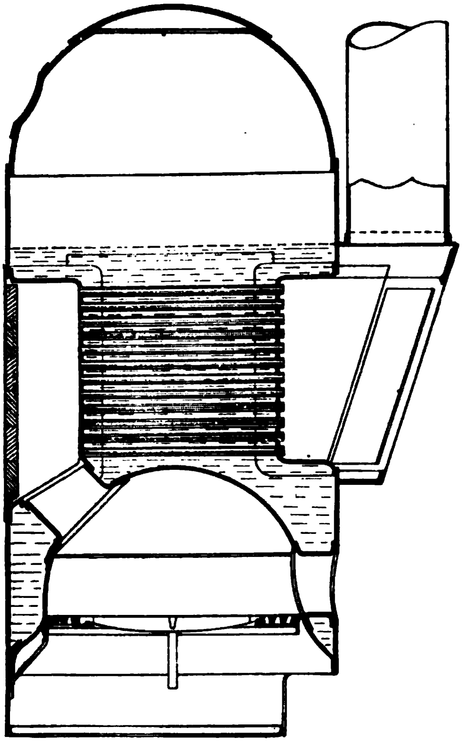


Fig. 35.

This boiler has a large combustion chamber; the fire box is surrounded by water, and the crown sheet and tubes are removed from the intense heat of the fire. This arrangement increases the heating surface, allows complete combustion and results in a durable boiler. The base is partially filled with water so that any sparks carried over will be quenched.

Robb-Mumford. This boiler resembles the through-tube internally-fired boiler (see Fig. 27) in that the fire is within a corrugated flue and the tubes lead from this flue to the rear of the boiler.

The Robb-Mumford boiler consists of two cylindrical drums or shells, connected at each end by a neck. See Fig 37. The upper drum is a steam and water drum, the water level being at about the middle. The lower shell is larger and is inclined about one inch in twelve to increase the circulation, and facilitate washing out. In this lower shell is the corrugated flue containing the grate. The fire tubes nearly fill the remaining portion of the shell as shown.

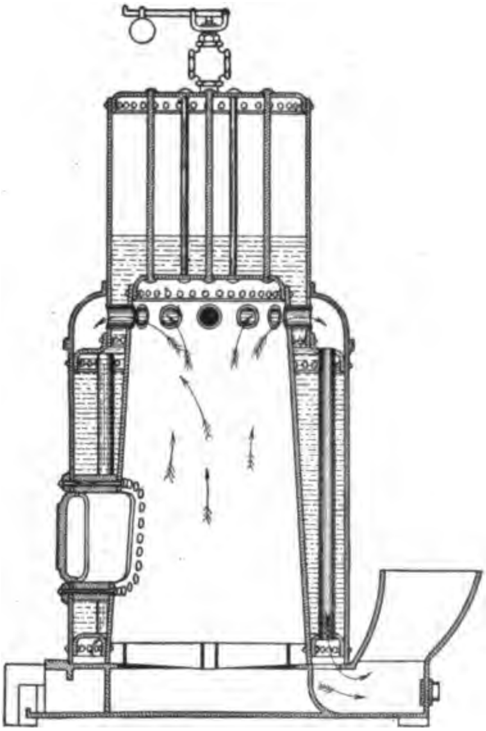
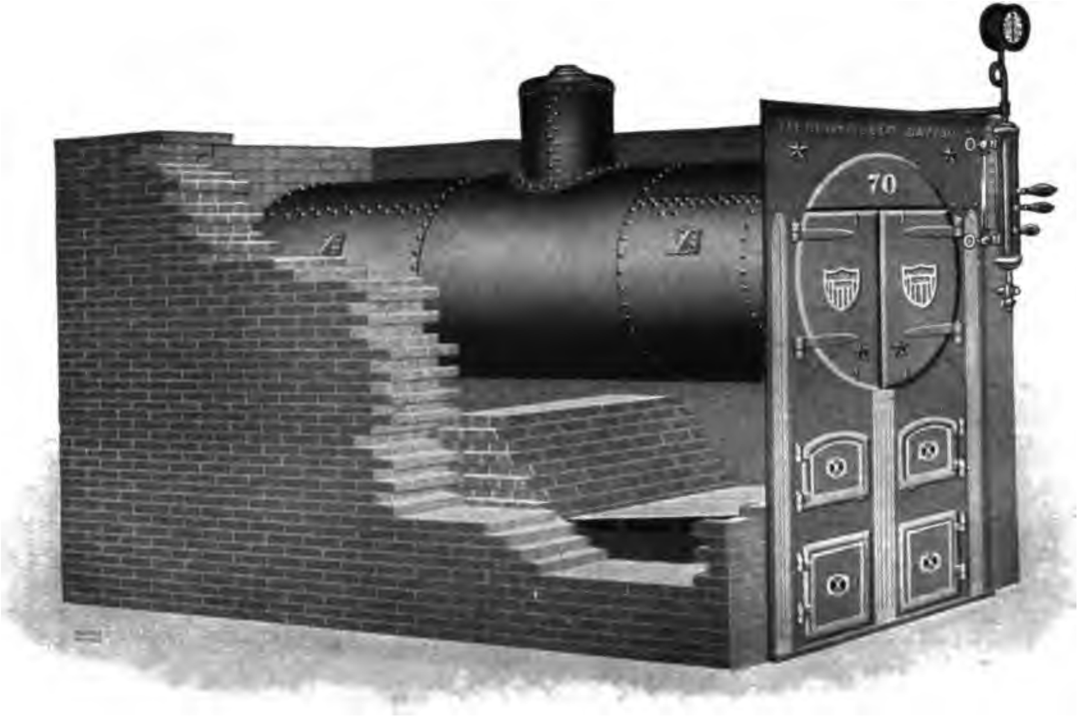


Fig. 36.

The furnace, in which the coal is burned, is surrounded by water; the hot gases pass through the tubes to the rear, return between the lower and upper shells and escape to the chimney from the front of the boiler. The steel casing keeps the gases in contact with the drums. The water circulation is shown by the arrows. The mixture of water and steam enters the upper drum; the steam here separates from the water which flows down the neck at the forward end. A semicircular baffle plate directs this water around the furnace to the bottom. The feed water enters at the rear of the steam and water drum.

As compared with the multitubular boiler one can readily see that the drums can be made much thinner on account of the small diameter. The tubes are short, straight, and easily cleaned. The internal furnace does away with much of the loss from leakage. As the water is well subdivided, steam can be raised rapidly and there is little danger of a disastrous explosion.



MULTITUBULAR BOILER.
The Brownell Company.

Directurn. The Begg's "Directurn" boiler (Fig. 38) is, in brief, a horizontal, externally-fired, multitubular boiler in which tubes conduct hot gases through the space behind the bridge.

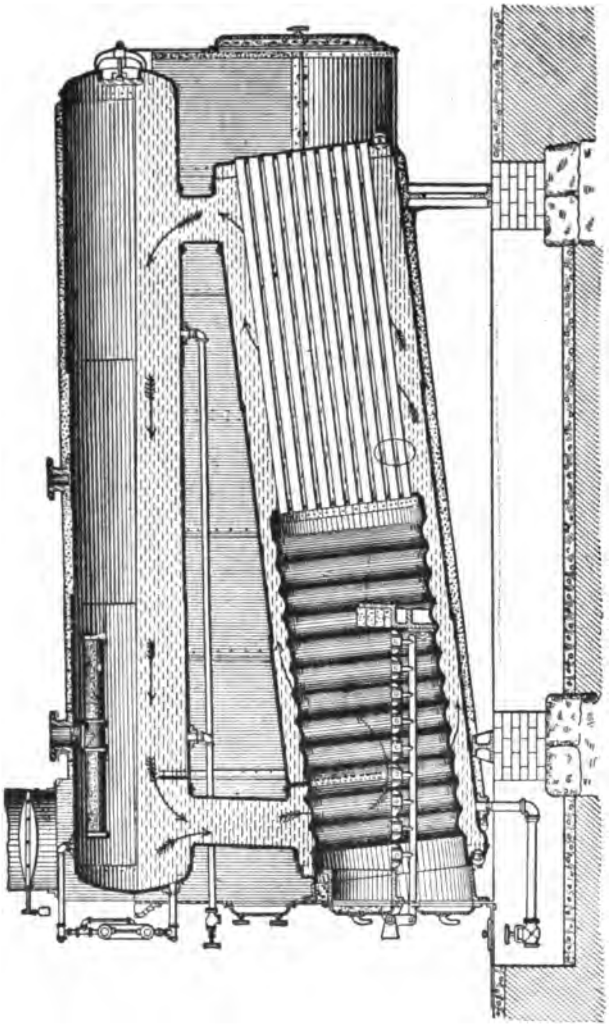


Fig. 37.

This boiler consists of a shell partly filled with 3-inch tubes. The rear of the furnace is a throat sheet in which 4-inch tubes are expanded. The other ends are expanded into the rear end plate which is made large enough for the purpose. The boiler is encased in steel plates lined with fire brick which is held in place by rods passing through the notches as shown. The manhole for entering the boiler is placed in the front head instead of in the shell as is frequently done.

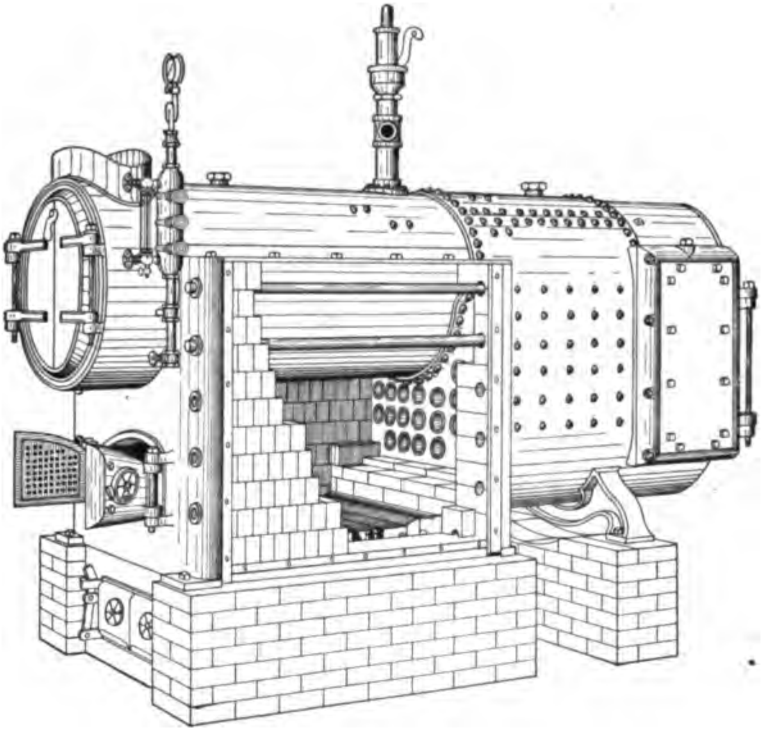


Fig. 38.

WATER-TUBE BOILERS.

The water-tube boiler differs essentially from the fire-tube. The names indicate the chief point of difference. In the fire-tube boiler, the tubes, which are surrounded with water, conduct the hot gases to the smoke box. In the water-tube, the tubes are filled with water, and the hot gases pass over and among them on their way to the chimney.

Although flue boilers and the tubular types were introduced at an earlier period than the water-tube, yet the last-named type is not a new form of steam generator. About a century ago, John Stevens invented a water-tube boiler and fitted it to a steamboat.

This boiler (Fig. 39) was a combination of small tubes connected at one end to a reservoir. Thus the "porcupine" was one of the earliest forms. At various times since then, many ideas have been worked out both for marine and stationary boilers. During the last fifteen years, however, the water-tube boiler has been steadily growing in favor, the chief reasons being—the necessity of higher steam pressures, greater reliability of materials, greater skill in design and workmanship, and more intelligent management.

It is not within the province of this instruction paper to discuss the relative merits of fire-tube and water-tube boilers, but a careful, impartial consideration seems to show that as far as economy of running is concerned there is but little difference. The fire-tube boiler is reliable and can be handled by those possessing comparatively little knowledge of engineering. Its chief defect seems to be the disastrous results following an explosion. The water-tube boiler, on the other hand, is safe, and suited to higher pressures, but requires greater care in management.

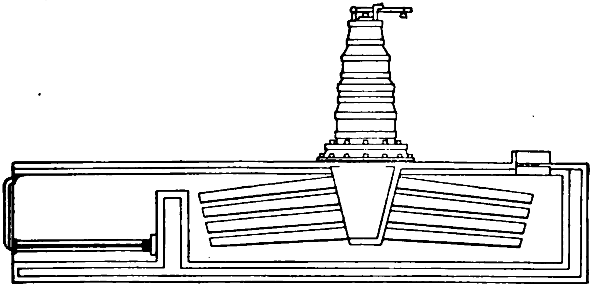


Fig. 39. Stevens Boiler.

Before discussing these boilers in detail, let us consider briefly the salient points.

Safety. Probably the greatest advantage claimed for the water-tube boiler is its safety. The boiler contains much less water than does the flue or tubular boiler and the water is divided into small masses, thus minimizing serious results in case of rupture. On account of the shape and arrangement of parts, the circulation is usually good, and no part exposed to the fire can be uncovered while there is any water in the boiler. The tubes cannot become overheated until the boiler is empty and with an empty boiler there cannot be a serious explosion.

Rapidity in Raising Steam. The many small streams into which the water is divided as it passes through the furnace greatly facilitate the absorption of heat. Because of the small streams and the rapid circulation, the water is converted into steam in a very short time. Several hours (usually five to seven) are required to raise steam to working pressure in a tubular boiler, while in many water-tube boilers, steam can be raised to over 200 pounds pressure in less than half an hour.

Durability. Most water-tube boilers are so designed that no seams are exposed to the fire or hot gases. The seams are the weakest part of a boiler, and as strains due to unequal expansion concentrate at such points, leaks or even ruptures are liable to occur. In the water-tube boiler, the joints between tubes and tube sheets are not in the direct path of the hot gases.

Loss of Heat. The loss of heat will evidently be reduced to a minimum if the heating surfaces are such that the heat readily passes through to the water. The small diameter of the water tubes (2 to 4 inches) allows the use of thin metal which does not hinder the transmission of heat. The rapid circulation in the water-tube boiler prevents the accumulation of sediment which is a poor conductor of heat. Still further, dust and dirt does not readily collect on the convex surface of water tubes, but the *inside* of fire tubes soon become choked with soot unless cleaned frequently. See Fig. 40.

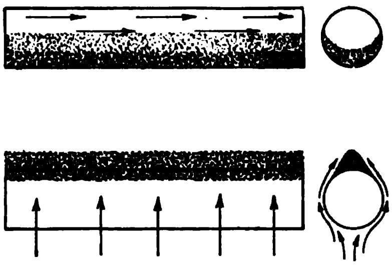


Fig. 40.

Less Weight. It is a well-known fact that a cylinder of large diameter must be much thicker than one of small diameter when the internal pressure is the same. The thickness of the shell of a fire-tube stationary boiler is not excessive, because of the moderate diameter; but in the return-tube marine boiler, the shell plates for 250 pounds pressure would be about 1⅞ inches thick. The difficulty of working such thick plates and their great weight render the cylindrical boiler unsuitable for high pressures. The small tubes and drums of the water-tube boiler may be made quite thin even for very high pressures. In general, it may be said that for the same capacity and pressure, the weight of a water-tube boiler is only about two-thirds that of a fire-tube.

CLASSIFICATIONS.

Many attempts have been made to classify water-tube boilers. By some writers a classification based on circulation, or on the principle of operation, is claimed to be superior to any division according to construction. Therefore, they divide them into classes as follows—boilers with limited circulation; boilers with free circulation; boilers with accelerated circulation.

In the first part of this Instruction Paper, is given a classification according to features of construction. No classification is altogether satisfactory because boilers overlap into other divisions; a water-tube boiler may be sectional, of the double-tube type, have horizontal tubes, straight tubes, and free circulation. In order to have some sort of classification, and as no discussion will be entered into regarding relative merits, the classification given on page 6 will be here adopted and followed as closely as conditions will permit.

Water-tube boilers are divided into two great classes—horizontal and vertical. Under these heads come sectional and nonsectional, straight-tube and curved-tube, and single-tube and double-tube. If the tubes are nearly horizontal, such as is the case of the Babcock and Wilcox, Root, etc., the boiler will be called horizontal. If the tubes are vertical, or nearly so, as in the Wickes, Stirling, etc., the boiler will be classed as vertical.

Although most boilers can be classified as outlined on page 6, there are a few of such peculiar construction and arrangement that they must be placed by themselves under "Peculiar Forms." These are described without any further attempt at classification.

As it is impossible to discuss all makes of boilers, a few representative forms will be considered as types of their respective classes. No attempt will be made to choose any make as being the best, because many conditions must be considered in selecting a boiler. The boilers described, except in a few cases, are now used extensively in either stationary or marine work.

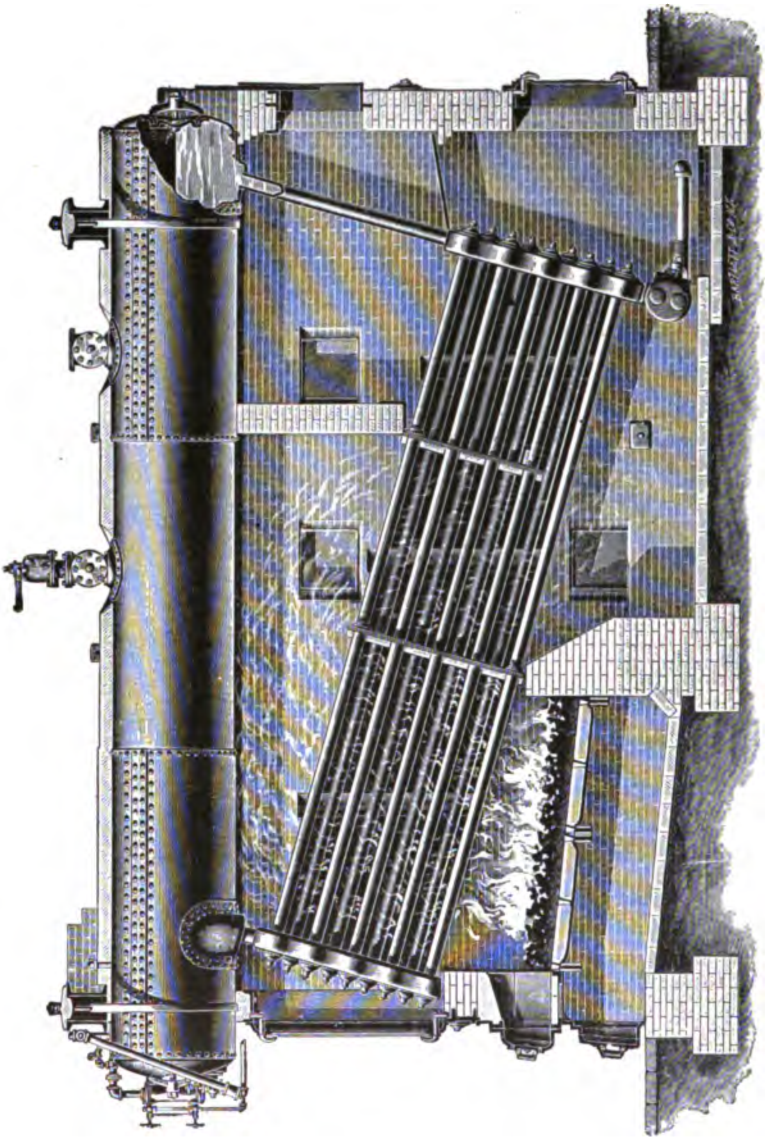


Fig. 41.

HORIZONTAL WATER-TUBE BOILERS.

BABCOCK AND WILCOX.

Water Tubes Nearly Horizontal—Steam and Water Drum Horizontal—Straight-Tube—Single-Tube—Sectional.

Construction. This boiler consists of a large number of lap-welded, wrought-iron, 4-inch tubes connected to each other and to a horizontal steam and water drum. The arrangement of the parts is shown in Fig. 41 which is a side view of a much-used form of this boiler. Each tube is expanded into a forging of the form shown in Fig. 42.

The tubes in a vertical row enter one piece and this vertical row is independent of the others, as shown in Fig. 43. Thus it is readily seen that this is a sectional boiler. Fig. 43 shows also the "staggered" arrangement of the tubes. In the back side of the front header, and in the front side of the rear header, holes are drilled into which are expanded the water tubes. In the front side of the header a flanged hole opposite each tube is fitted with a hand-hole plate. The details of construction are shown in Fig. 44. The tops of the headers are connected to the steam and water drum by short tubes and the same construction is used for connecting the mud drum to the rear header.

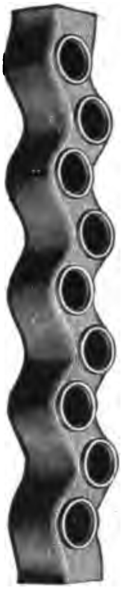


Fig. 42.

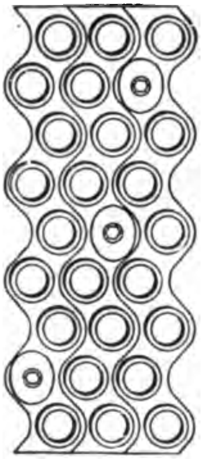


Fig. 43.

Operation. The grate is at the front end of the boiler under the higher end of the tubes. The hot gases from the fire are guided by division plates and bridges, so that after rising from the grate they pass between the tubes to the combustion chamber, which is under the steam and water drum; the gases then pass downward among the tubes, and after rising a second time pass off to the chimney. In this way, the direction of the currents of hot gases is at all times almost at right angles to the tubes, thus impinging upon them instead of passing parallel to the heating surfaces, as in the case of fire tubes. As the gases impinge three times against the staggered tubes, the heating surface is very efficient.

Circulation. The feed water enters the steam and water drum through the pipe shown in Fig. 44. It is thus heated before it mixes with the hot water in the boiler. As the water in the tubes becomes heated, it rises to the higher end where it is partly converted into steam; a column of water and steam rises through the header to the drum in which the steam and water become separated. The cooler water at the rear of the steam and water drum flows down into the lower end of the tubes and as it becomes heated rises. Thus there is a continuous circulation.

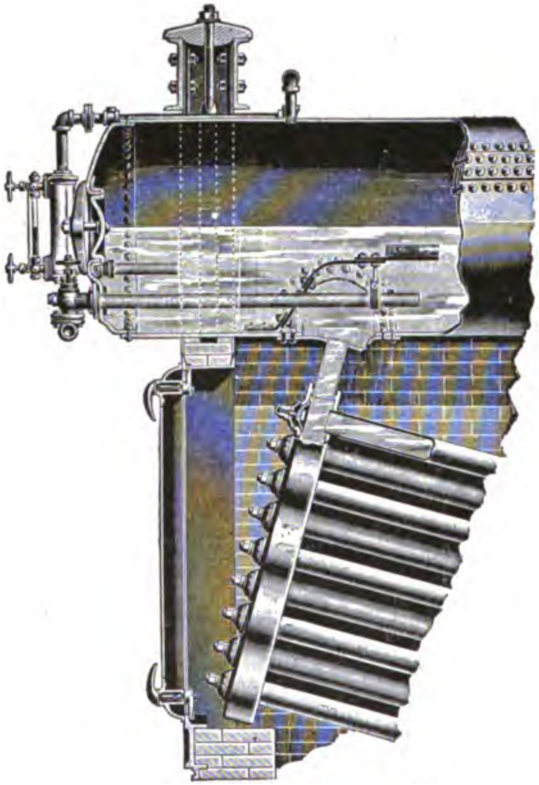


Fig. 44.

Steam is taken from the rear end of the steam and water drum. The solid matter in the water is not deposited on the tubes because of the rapid circulation; it falls to the mud drum from which it is blown out.

The marine form of this boiler has a cross drum, that is, the drum is at right angles to the tubes instead of parallel to them. It is similar in form to the cross-drum types used for stationary work. This form is used in case there is not sufficient head room.

ROOT.

Water Tubes Nearly Horizontal—Steam and Water Drums Horizontal —Straight-Tube—Single-Tube—Sectional.

The above brief outline indicates that the Root water-tube boiler is, in its main features, like the Babcock and Wilcox. In fact the difference is in detail of construction only. Fig. 46 shows the general appearance when a part of the brickwork is removed. It will be seen that there is a large steam drum (cross type) at the top in addition to the small steam and water drum over each section.

Construction. The Root water-tube boiler is composed of 4-inch lap-welded wrought-iron tubes. These tubes are expanded into cast iron headers as shown in A, Fig. 45. A vertical section is formed by placing one pair upon another as shown at B, Fig. 45. One tube of each pair is connected to one above it by a flexible bend, by means of which is obtained an uninterrupted circulation from the bottom to the top of the section. A metallic packing ring (see C, D, and E, Fig. 45) insures a tight joint between the bend and the header. F, Fig. 45, shows an enlarged end of a bend.

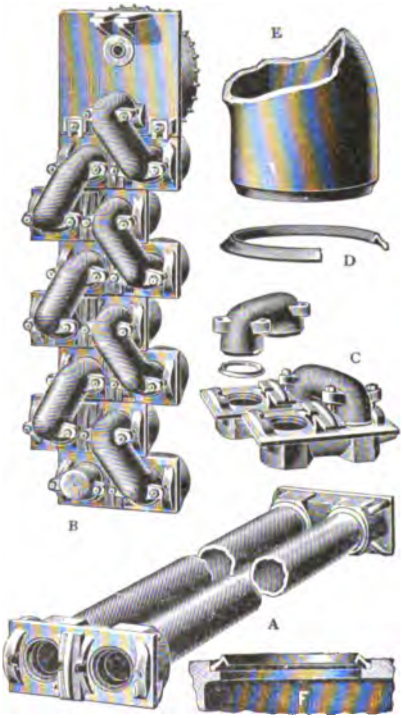


Fig. 45

To form the boiler several of these vertical sections are placed side by side. These vertical rows are not rigidly connected because the lower tubes being nearer the fire expand more than those above.

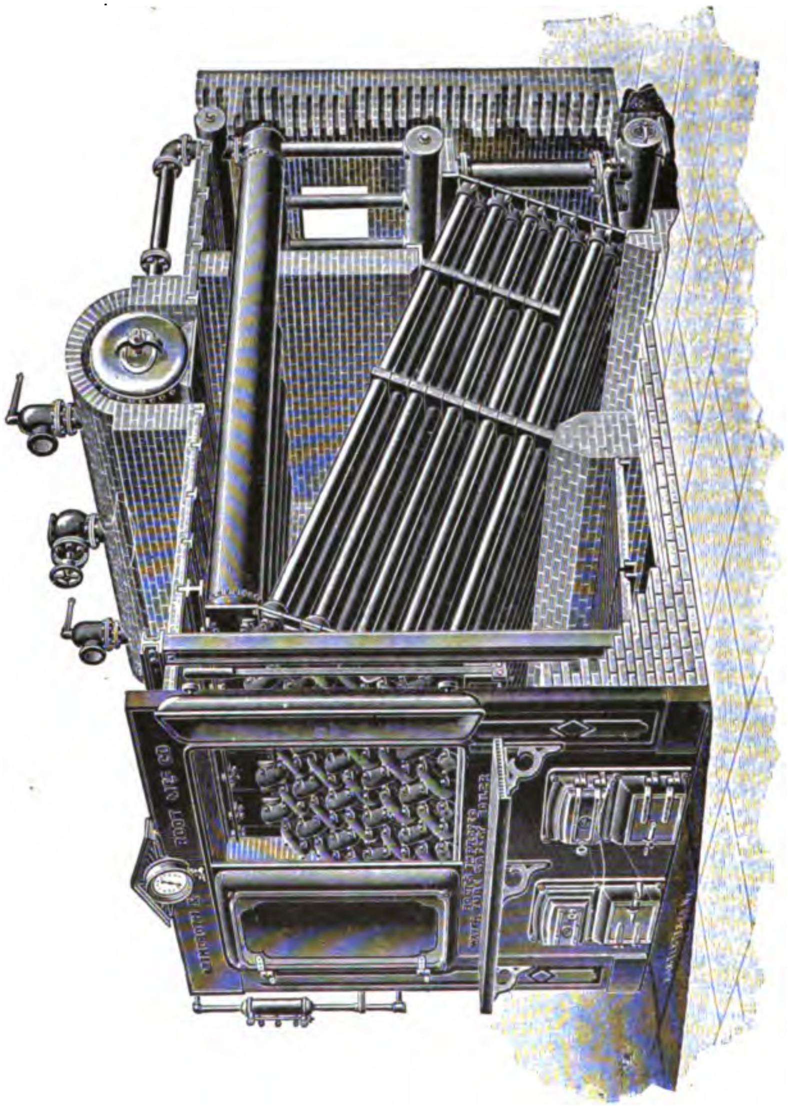


Fig. 46.

Circulation. Each section has its overhead drum into which the water and steam is discharged from the tubes. At the rear of the boiler and at the end of each steam and water drum, a vertical pipe leads to a cross drum beneath; this drum is a common reservoir for all the sections. The feed water enters this drum and meets the hot water coming from above. The mixing of the water results in a temperature which prevents any trouble from unequal expansion. The cross drum (reservoir) is also connected by vertical pipes to another drum which is below and parallel to it; this is the mud drum. From the feed reservoir, the mixture of feed and circulating water descends to the mud drum in which the solid impurities are left. The circulating water then flows from the top of the mud drum into the lower end of the tubes. As these tubes are surrounded by hot gases, the water becomes heated and rises through the tubes to the steam and water drums. This heated water contains bubbles of steam which leave the water and collect in the steam drum. The water flows through the steam and water drum and descends to meet the entering feed water. The water level is at about the middle of the steam and water drums.

The hot gases from the fire pass among the tubes three times in practically the same manner as in the Babcock and Wilcox boiler.

WORTHINGTON.

Water Tubes Nearly Horizontal—Steam and Water Drum Horizontal —Straight-Tube—Single-Tube—Sectional.

Construction. This form of boiler is much the same in principle and operation as the Babcock & Wilcox boiler, but the parts are differently proportioned and arranged; see Fig. 47. The furnace extends under the entire boiler, and the tubes are set over it close together in oppositely inclined series. No flame walls or baffle plates are used.

Boilers up to 125 H. P. are usually made to fire at the end as shown in Fig. 47, in which the tubes extend across the furnace viewed from the front, and the steam and water drum is at right angles to the tubes. In the *side-fired* boilers the tubes extend from front to back, and the steam and water drum from side to side; this arrangement is better adapted for large units and for setting in battery. The tubes of each vertical row are expanded into straight headers which contain seven or eight tubes. See Fig. 48. Opposite each tube is a hand hole. These headers are arranged close together, forming the boiler enclosure.

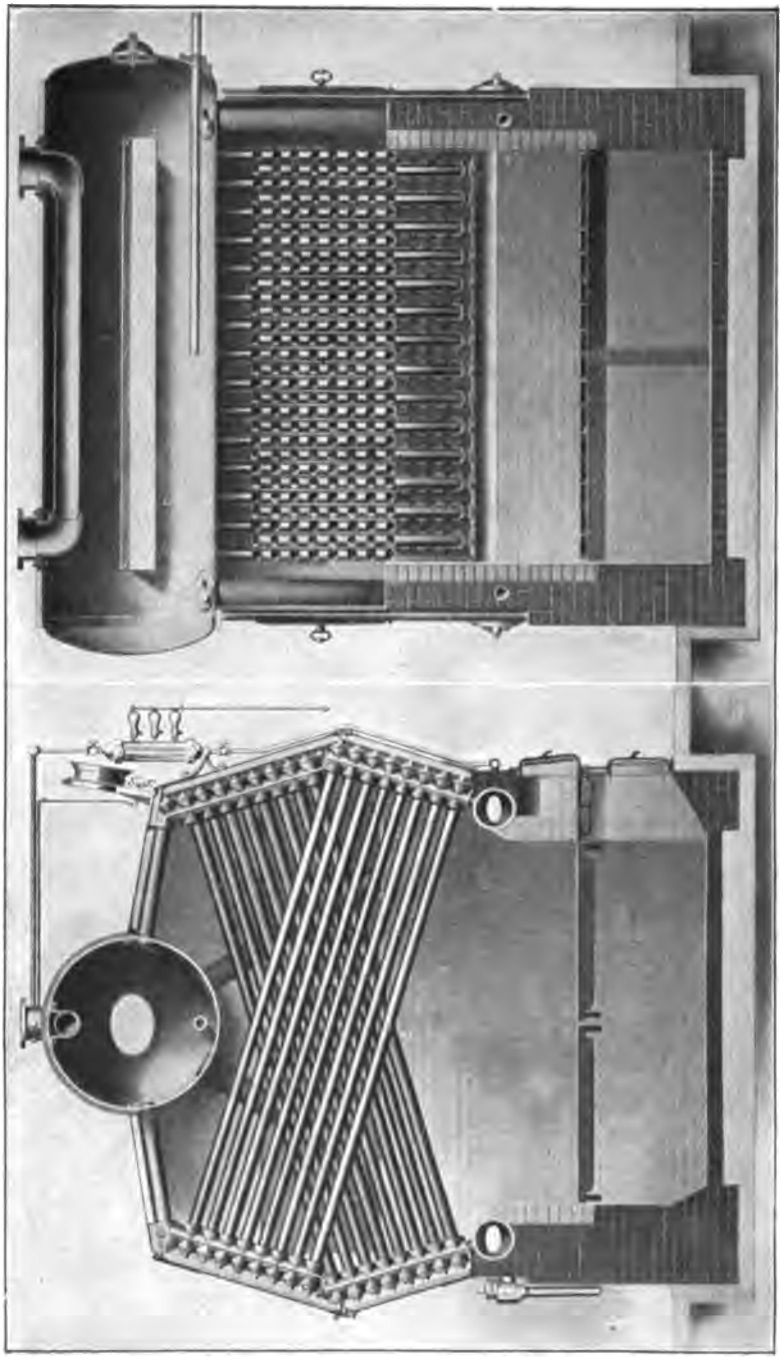


Fig. 47.

Circulation. The feed water enters the steam and water drum and the circulation carries it down to the mud drums through large circulating tubes which are outside of the furnace. See Fig. 48. From the mud drum it enters the lower series of headers and rises through the inclined tubes over the fire into the upper headers.

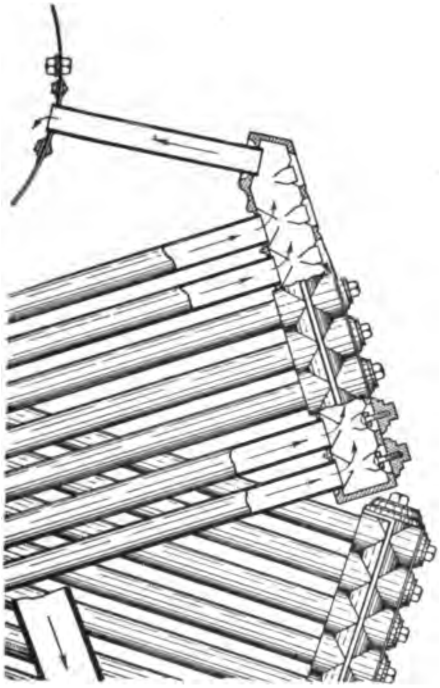


Fig. 48.

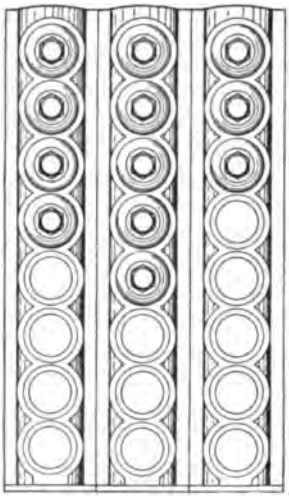


Fig. 49.

The water now containing bubbles of steam enters the steam and water drum by means of short tubes shown in Figs. 47 and 48.

The covering for this boiler is an iron casing, no brick being used except to enclose or line the furnace.

HEINE.

Water Tubes Nearly Horizontal—Steam and Water Drum Parallel to Tubes—Straight-Tube—Single-Tube—Non-Sectional.

Construction. The Heine water-tube boiler is not a sectional boiler. Instead of being expanded into small headers grouped to form a boiler, all the tubes are expanded into the inside plates of

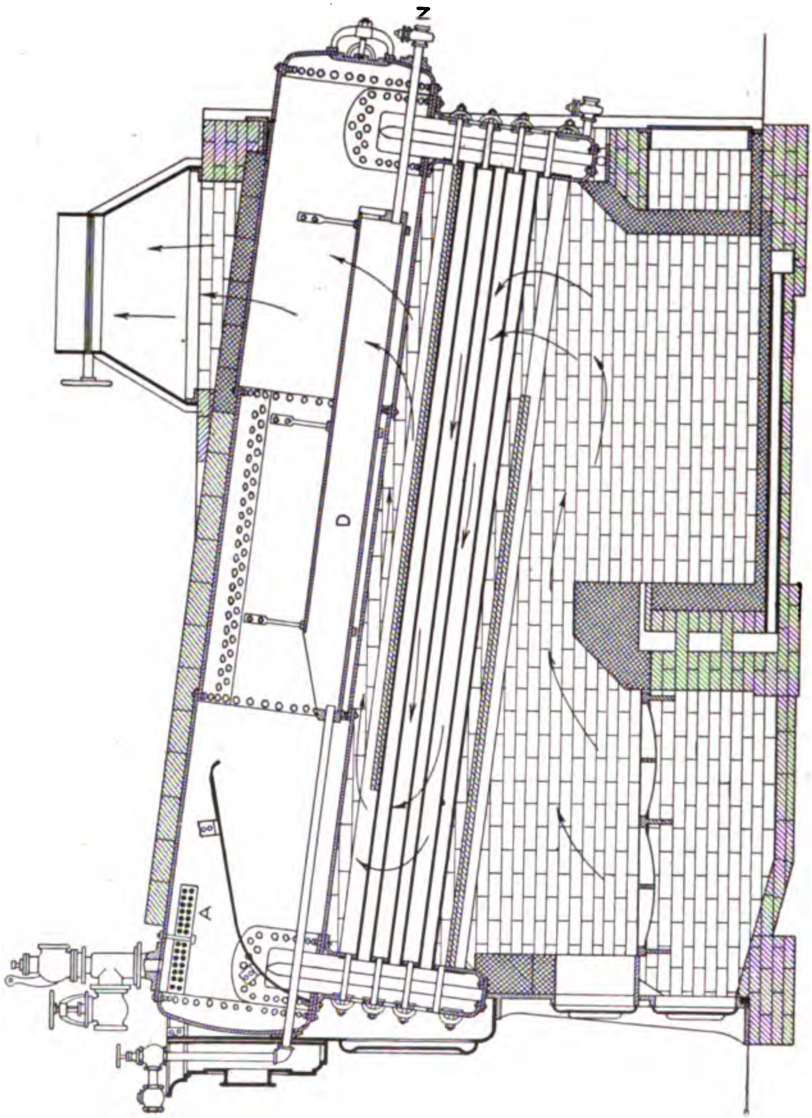


Fig. 51.

a water leg at each end. The construction of this water leg is shown in Fig. 50. It is composed of two parallel plates flanged and riveted to a butt strap. The plates are strengthened by short hollow screw stays similar to those used in the water-leg construction of fire-box boilers. At the top, the water leg is curved and joined to the steam and water drum by riveting. Opposite each tube is a hand hole for cleaning or replacing a defective tube.

Circulation. The feed water enters at the front of the steam and water drum and flows into the mud drum D, from which it passes to the rear header with much less velocity. The water is warmed while passing through the pipe leading to the mud drum, and as it flows slowly through the mud drum it deposits its sediment. The accumulated sediment is blown off by means of the blow-off pipe N. The water, as it becomes heated in the mud drum, rises and passes to the front of the mud drum, from which it flows in a thin sheet to the rear of the steam and water drum and to the rear water leg. From the rear water leg, it enters the tubes in which it is partially converted into steam. The mixture of steam and water enters the higher end of the drum from the water leg, and as there is but a thin layer of water in the steam and water drum, the steam readily rises through it. A deflection plate prevents water from being carried to the perforated steam pipe A.

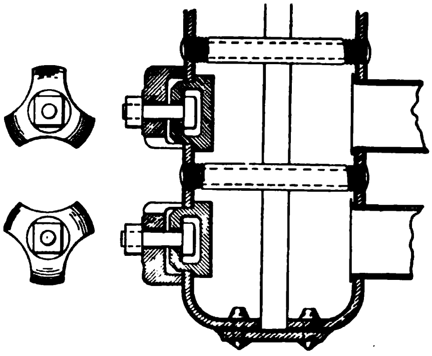


Fig. 50.

The flow of hot gases from the fire is directed by light tile placed on the upper and lower rows of tubes as shown in Fig. 51. The hot gases flow nearly parallel with the tubes instead of across them as in the Babcock and Wilcox.

ATLAS.

Water Tubes Nearly Horizontal—Steam and Water Drums (Cross Type) Horizontal—Straight-Tube—Single-Tube—Non-Sectional.

This make of water-tube boiler does not need a full description as Figs. 52 and 53 show both the general arrangement of

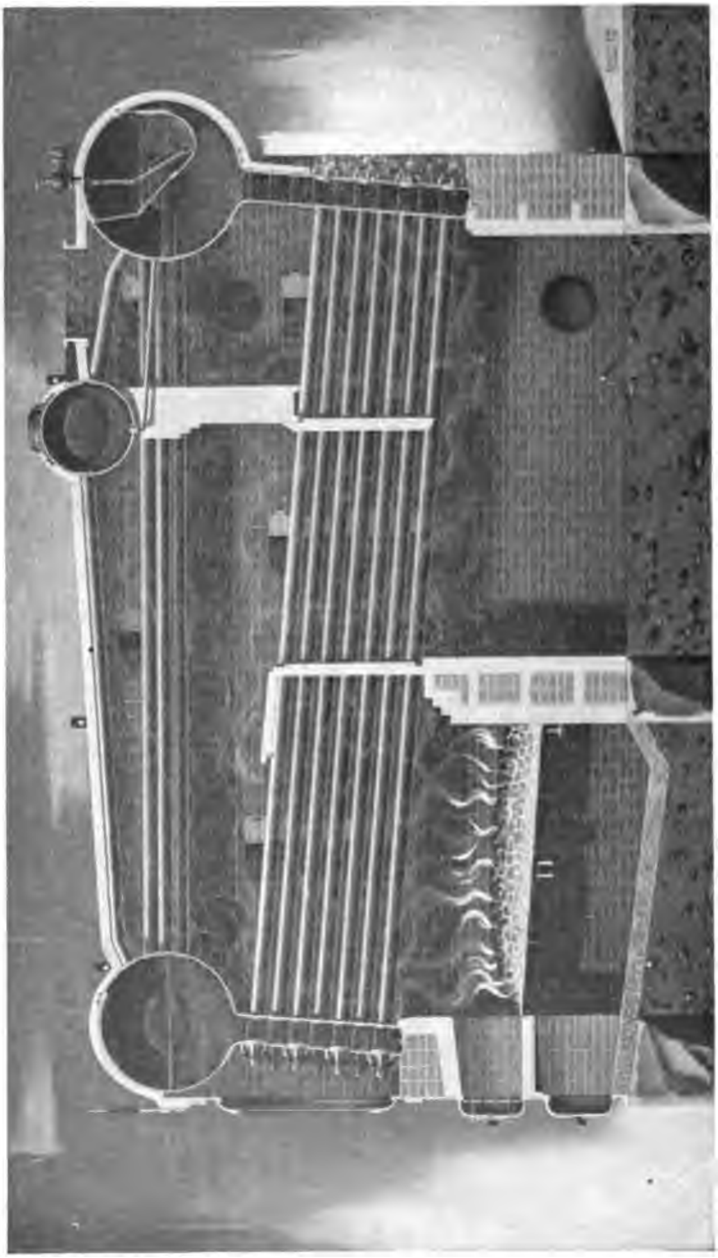
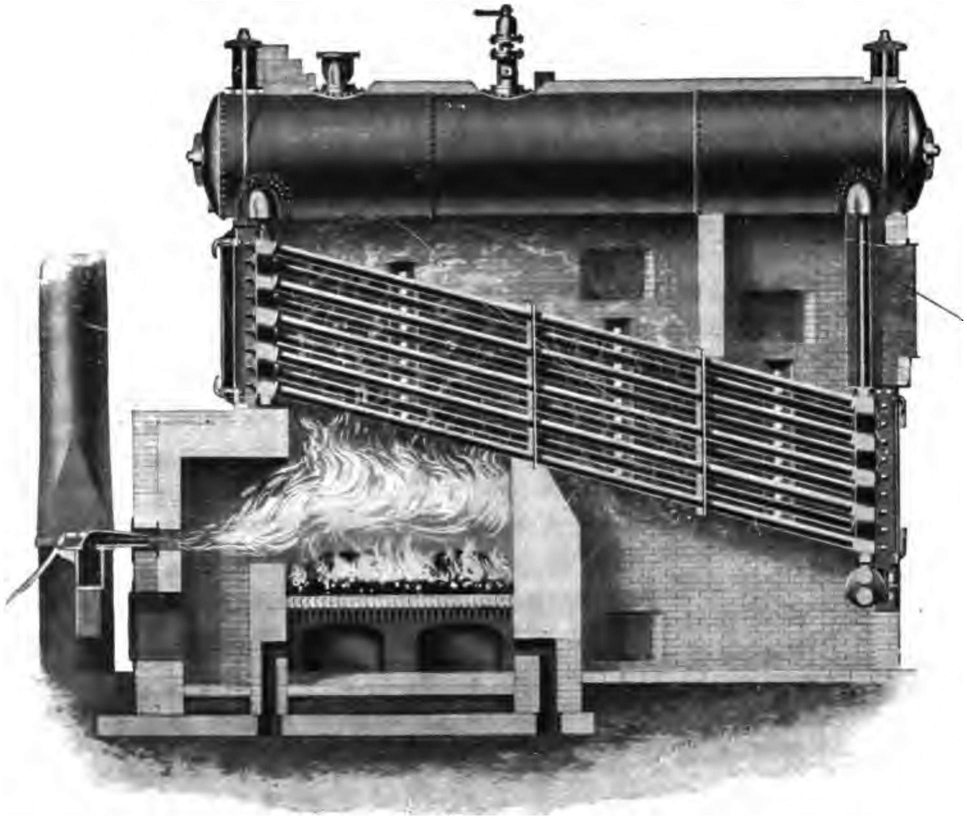


Fig 52.



Section of Babcock and Wilcox Boilers Fitted with Kennedy Burners for Burning Blast Furnace Gas.

parts and the details of water-leg construction. The general description of the Heine boiler is applicable to this type. A few points of difference, however, should be pointed out.

There are three drums running crosswise the tubes. The front and rear drums are made of the same plates as the water legs. This is shown in the illustrations. The reasons for this method of construction are that it gives a "throat" area of about 80 per cent of the area of the leg and prevents all seams from coming in contact with the furnace gases. The two drums are connected on the water line by equalizing tubes. In the rear drum, a water purifier receives the feed water which passes down the rear leg, then through the tubes to the front leg. In the last portion of the tubes, much of the water is converted into steam which flows through the front drum and the superheating tubes to the small upper drum. The water flows through the equalizing tubes to the rear drum and joins the current of feed water.

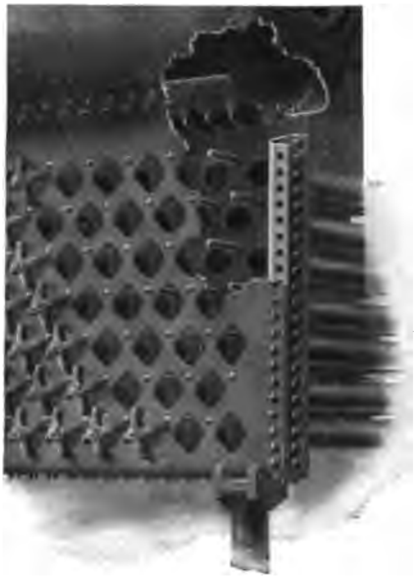


Fig. 53.

The front drum is made 36 inches in diameter, the middle drum 24 inches, and the rear drum 42 inches. The tubes are 4 inches in diameter and the longest are 18 feet in length.

MOSHER.

Water Tubes Nearly Horizontal—Steam and Water Drums (Cross Types) Horizontal—Curved-Tube—Single-Tube—Non-Sectional.

The chief differences in appearance between this boiler and those already described are shorter tubes, making a more compact boiler, and the curved tubes. This type is more often used in

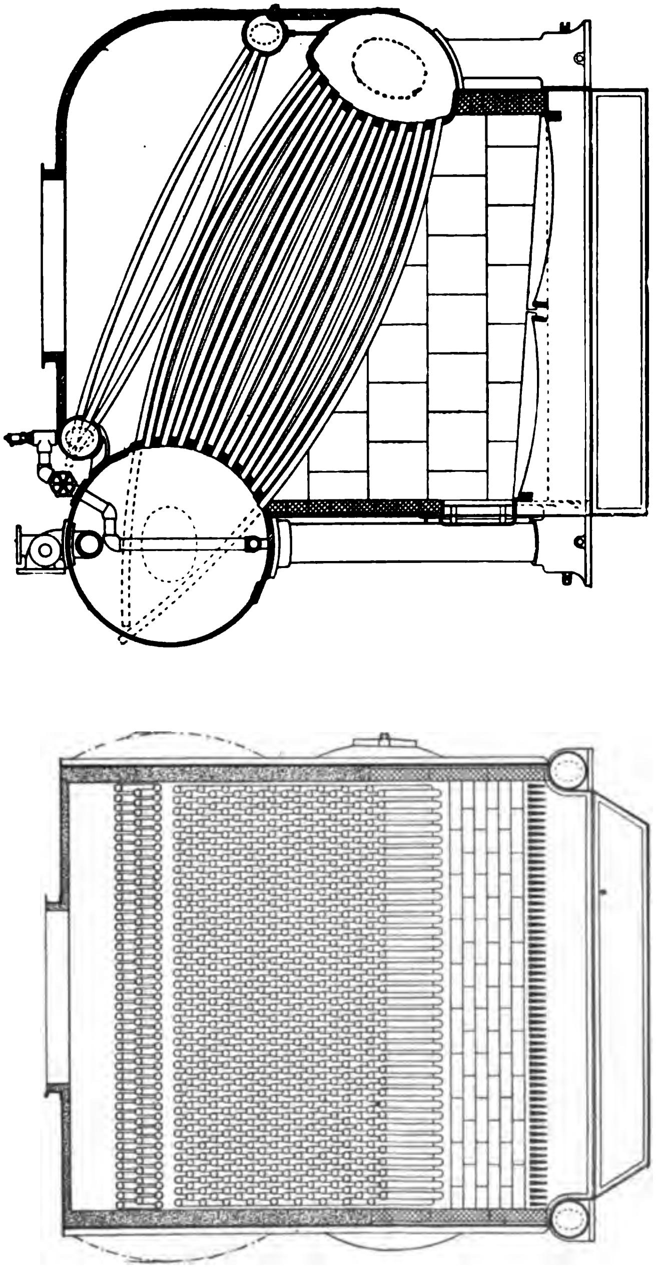


Fig. 54.

marine than in stationary work. The boiler consists of a large steam and water drum connected to a smaller water drum by slightly curved tubes. The steam drum is supported by two large circulating pipes (one at each end) which are connected by other pipes to the water drum. Thus the circulation is down these pipes and along the pipe at the bottom (see Fig. 55), up to the water drum and from thence to the steam drum by the tubes which are in contact with the hot gases.

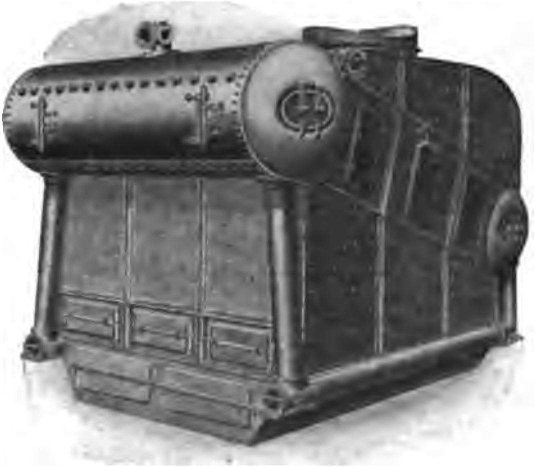


Fig. 55.

The feed-water heater, shown in Fig. 54, consists of two small drums connected by tubes. The parallel dotted lines in the steam drum of Fig. 54 show how tubes are removed and replaced. Fig. 55 shows the row of plugs for this purpose. These plugs are illustrated in Fig. 56. Each plug is a conical-headed bolt, having a short piece of copper tube, a washer, and a nut. The conical head and the copper tube are inserted in the hole until the washer is in contact with the outer surface of the drum. The nut is then screwed up, thereby flaring the end of the copper tubing as shown.

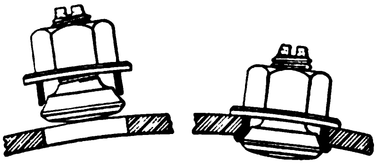


Fig. 56.

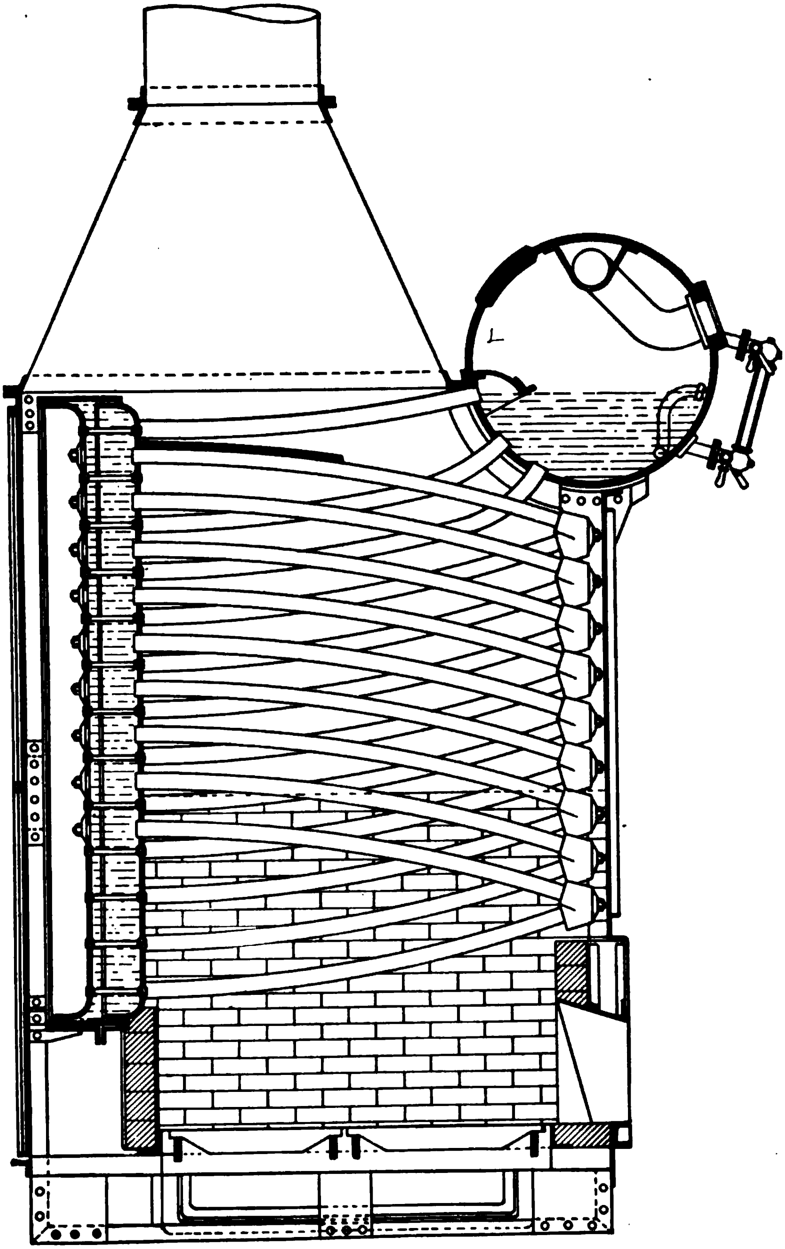


Fig. 57.

The steam pressure on the conical head increases the tightness of the joint.

THORNYCROFT-MARSHALL.

Water Tubes Nearly Horizontal—Steam and Water Drum (Cross Type) Horizontal—Curved-Tube—Single-Tube—Non-Sectional.

The Thornycroft-Marshall non-sectional boiler consists of a large horizontal steam and water drum, a vertical water box or header, and the generating tubes. Like the Mosher, the tubes are curved slightly, but the header is a distinct difference.

The general features of construction are shown in Fig. 57. The steam and water drum, sometimes called the separator barrel, is simply a cylinder with dished ends. The water level is about one-third the diameter of the cylinder. The tubes, which are 3¼ inches in diameter, are connected in pairs to a junction box at one end and to a water box or header at the other end. Thus each pair forms a unit, but the two tubes of the unit are not in the same vertical plane. The upper tube enters the header as high as possible and the lower ones enter low down, thus giving considerable upward slope. From near the top of the water box, three rows of tubes lead to the separator barrel as shown in Fig. 58. The water box is very simple, the flat plates are stayed by short hollow screw stay-bolts. The junction boxes are not restrained in any way; this construction, combined with the slight curve of the tubes, allows free expansion. The slight curve also allows the tubes to enter the separator barrel and the water box at right angles so that they may be expanded in place.

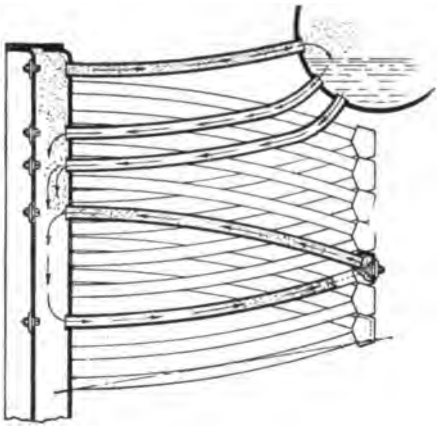


Fig. 58.

Circulation. The feed water enters the steam and water drum and then passes to the water box through the two lower sets of tubes. See Fig. 58. The water enters the lower ends of the various pairs

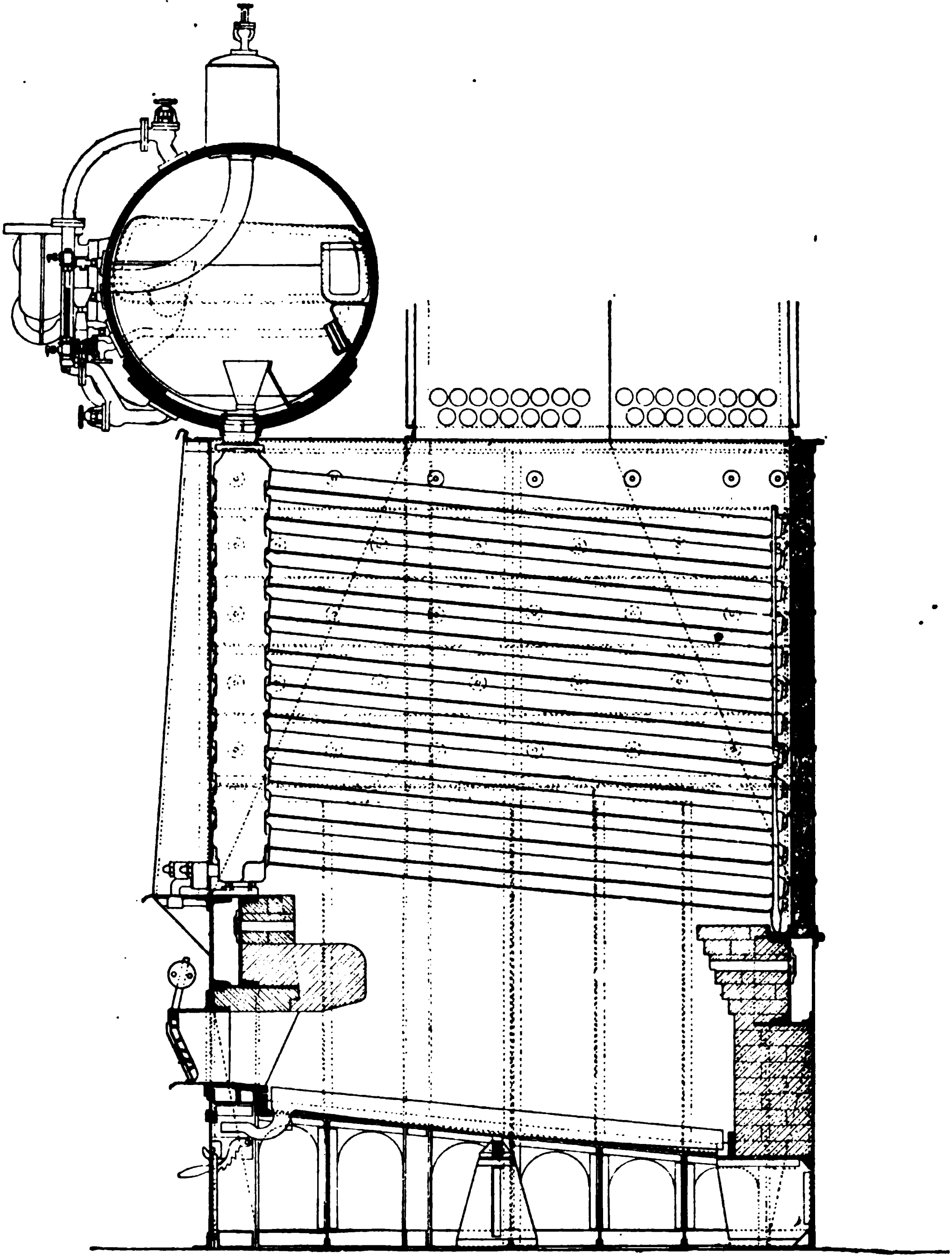


Fig. 59.

of tubes, as shown in Fig. 58, and rises in the tubes while in contact with the hot gases from the furnace. The mixture of steam and hot water then enters the header from which it passes to the steam and water drum by means of the highest row of tubes. The difference in height of the two tubes of a unit insures good circulation. A baffle plate prevents the water from splashing to the steam pipe.

The hot gases pass upward among the tubes which cross so frequently that they take almost all the heat from them.

NICLAUSSE.

Water Tubes Nearly Horizontal—Steam Drum Horizontal—Straight-Tube—Double-Tube—Sectional.

This boiler differs essentially from those already described in that it is of the *double-tube* type. In general, it consists of a number of elements which form a vertical header, to which tubes are connected. The tubes are set at an angle of about 6 degrees to the horizontal. Above the elements is a transverse steam and water drum which is in communication with the headers. The general arrangement of parts is shown in Fig. 59.

Construction. The interesting features of this type of boiler are the design and construction of the tubes and headers. To increase the circulation the principle of the "Field" tube is employed. In this construction, the outer, or generating, tubes ($3\frac{1}{4}$ inches in diameter) are closed at one end. Each generating tube contains an inner circulating tube which is $1\frac{19}{32}$ inches in diameter.

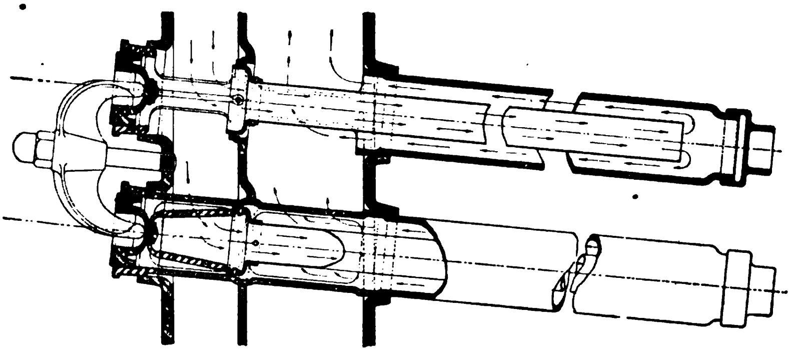


Fig. 60.

This tube is open at both ends. The closed ends of the generating tubes are supported by resting in holes in a plate or rack at the rear of the boiler. The forward end of the circulating tube is attached to a cap which screws into the outer end of the generating tube. A recess in this cap provides a bearing for an arch bar which spans two tubes, keeps them in place, and is itself secured by a nut on a bolt which is screwed into the header. See Fig. 60.

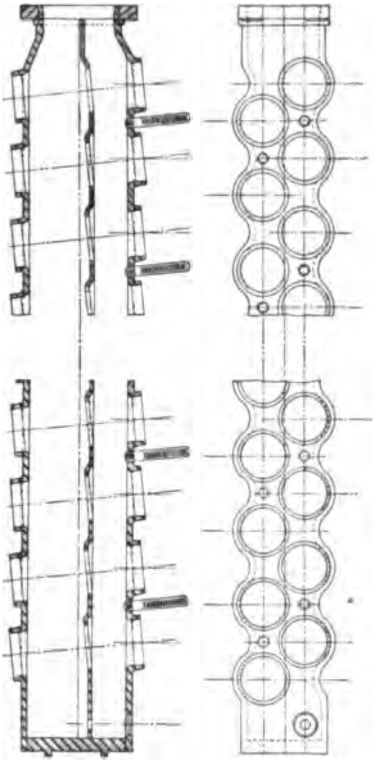


Fig. 61.

The front end of the generating tubes is of peculiar shape. To allow the water to enter the circulating tubes, and to fasten the tubes to the header without expanding them, each generating tube is provided at the open end with two cone-shaped portions; these are about eight inches apart. The first cone fits into a taper hole flanged outward in the front face of the header, and the second cone fits a similar hole in the rear face of the header. Both the holes and tubes are ground to the same size and taper. About midway between the cones, a third expanded portion occupies the tube hole in the diaphragm or middle plate of the header. See Fig. 60. The portion of the tube within the header is called the "lantern". At this point the tube is cut away so that water may freely enter the tube, the openings being above and below. In Fig. 60, the upper tube is in its normal position, but the lower tube has been turned through 90 degrees to show the construction.

To stand high steam pressures, the elements of the headers are made of wrought steel and are sinuous in shape. Fig. 61 shows the shape of the header and the positions of the tubes.

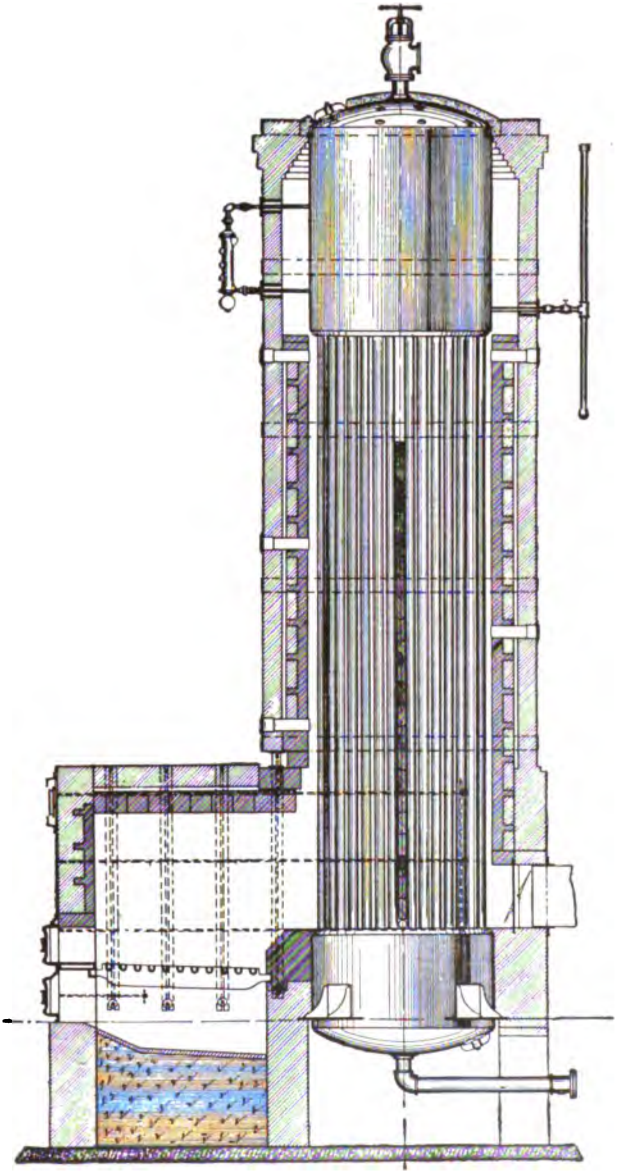


Fig. 62.

Each element contains 24 tubes in two vertical rows of 12 each In the middle of the headers, there is a diaphragm for dividing the interior. The front passage serves as a "downcomer" for the water, and the rear is the "upcomer", or riser, for the mixture of steam and water.

The lower ends of the headers are closed, and the upper ends flanged to connect with the steam and water drum, which is 42 inches in diameter.

Circulation. Fig. 60 gives an idea of the direction of circulation. Water from the drum descends in the front compartment of the header, flows into the circulating tubes, *which communicate with the front compartment only*, and after flowing the length of the circulating tubes, enters the generating tubes. The water then comes back through the annular spaces in the generating tubes to the rear compartment of the header, because the *generating tubes communicate with the rear compartment only;* while in the annular space it is partially evaporated. The mixture of steam and water then rises to the drum.

VERTICAL WATER-TUBE BOILERS.

WICKES.

Water Tubes Vertical—Straight-Tube—Single-Tube—Non-Sectional.

Let us now consider a water-tube boiler having vertical tubes. Fig. 62 shows the general arrangement of the parts of the Wickes vertical water-tube boiler. At the top is a cylindrical steam and water drum into which the upper ends of the vertical tubes are expanded. At the bottom is a cylindrical mud drum of the same diameter as the upper drum. The tubes are straight and plumb when in position; they are arranged in parallel rows with a clear space between rows to admit a small hoe to remove any soot that may accumulate on the tube sheet of the mud drum.

The tubes are divided into two compartments by heavy fire-brick tile. The tubes in the section next the furnace are called "risers"; those in the rear are the "downcomers," because the heated water rises to the steam drum through the front tubes, and the cooler water flows down those in the rear. The feed water is introduced into the upper drum. The direction of flow of hot

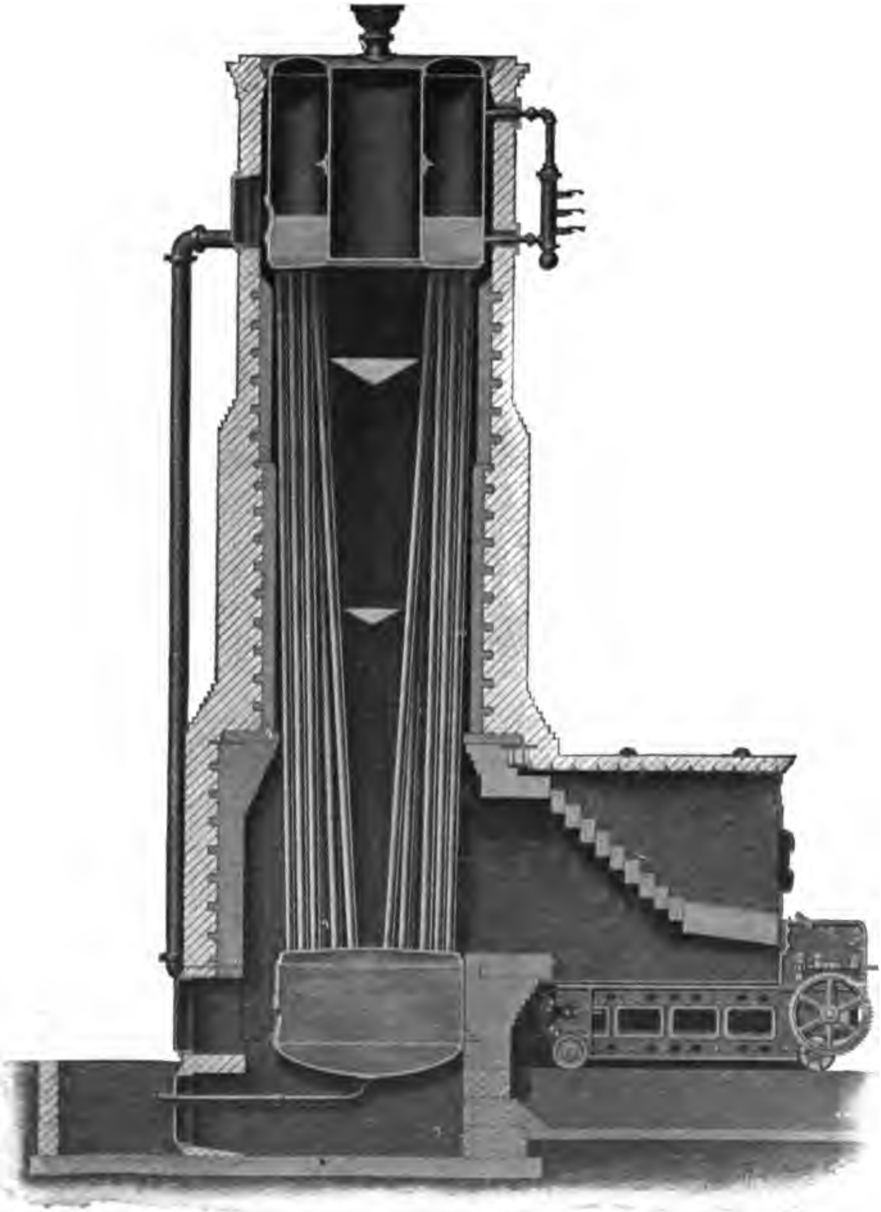


Fig. 63.

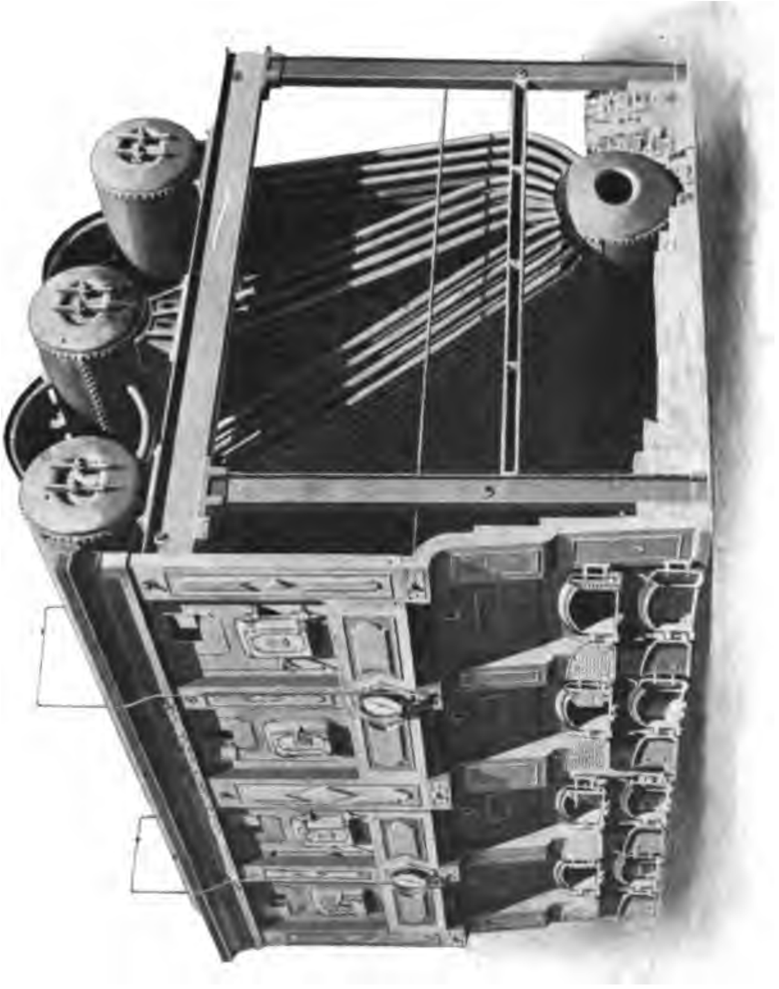


Fig. 64.

gases is the same as that of the water. A baffle plate in the steam and water drum directs the water to the downcomers.

The furnace is external and built entirely of brick. The hot gases from the fire come in contact with the tubes without passing through a combustion chamber.

CAHALL.

Annular Steam and Water Drum—Water Tubes Vertical—Straight-Tube—Single-Tube—Non-Sectional.

The Cahall vertical water-tube boiler consists of an annular steam and water drum, a cylindrical mud drum, and 4-inch vertical tubes. The generating tubes connect these two drums and are placed within the brick setting. An external circulating pipe also connects the two drums. As this pipe is filled with comparatively cool water and the generating tubes with a mixture of hot water and steam, the circulation is positive and rapid. The feed water enters the steam and water drum, flows down the external pipe to the mud drum and then rises in the generating tubes to the steam and water drum.

The fire is in a brick furnace at one side of the boiler as shown in Fig. 63. The hot gases rise among the tubes. The annular form of the steam drum makes the central space conical; in this space several deflecting plates, or baffles, cause the hot gases to flow out among the tubes. After heating the water in the tubes the hot gases pass through the opening in the steam and water drum coming in contact with the metal containing the steam This thoroughly dries the steam and in many cases slightly super heats it.

The steam drum and also the mud drum are equipped with swinging manheads. The steam drum also has several handholes for use in removing and replacing tubes.

STIRLING.

Water Tubes Nearly Vertical—Steam and Water Drums Horizontal—Curved-Tubes—Single-Tube—Non-Sectional.

The Stirling boiler, shown in Fig. 64, consists of three cylindrical steam and water drums at the top, and a mud drum at the bottom. The lower drum is connected to the upper drums by three sets of tubes which are curved slightly at the ends. The curved tubes allow for expansion and make it possible to have the tubes enter the drums radially.

The feed water enters the rear steam and water drum and coming in contact with the hot gases just before they enter the uptake, becomes gradually warmed. This heating causes most of the sediment to fall to the mud drum from which it may be blown out at intervals. The mud drum is protected from the intense heat of the furnace by the bridge wall.

Each set of tubes are separated from the others by partition walls or baffles of fire-brick tile so that the gases from the furnace pass along the entire length among the first set of tubes; they are then guided downward among the second set and after rising again among the tubes of the third set, escape to the chimney. By thus having a long passage a large proportion of the heat is taken from the gases before they go to the chimney. The fire-brick arch just above the furnace insures an even distribution of the gases and promotes combustion; the arch heats the entering air to a high temperature, thus reducing the liability of chilling the tubes by an inrush of cold air.

Steam is taken from the middle drum which is set a little higher than the others in order to obtain more steam space and drier steam. The boiler is surrounded on the rear and two sides by the brick setting; the front is of cast iron or of pressed steel. Numerous openings in the brickwork allow entrance for cleaning.

This type of boiler is flexible and adapted to cramped places as it can be made broad with little height or high with small floor area. All parts are either cylindrical or spherical in shape and of wrought metal. The curved tubes reduce the strains resulting from unequal expansion and contraction.

MILNE.

Water Tubes Vertical—Steam and Water Drum Horizontal—Curved-Tube—Single-Tube—Non-Sectional.

This boiler (Fig. 65) is in many respects similar to the Stirling (Fig. 64), but an inspection of the two illustrations will show several differences. In the Milne boiler there is but one steam and water drum and the tubes are vertical with a slight curve at

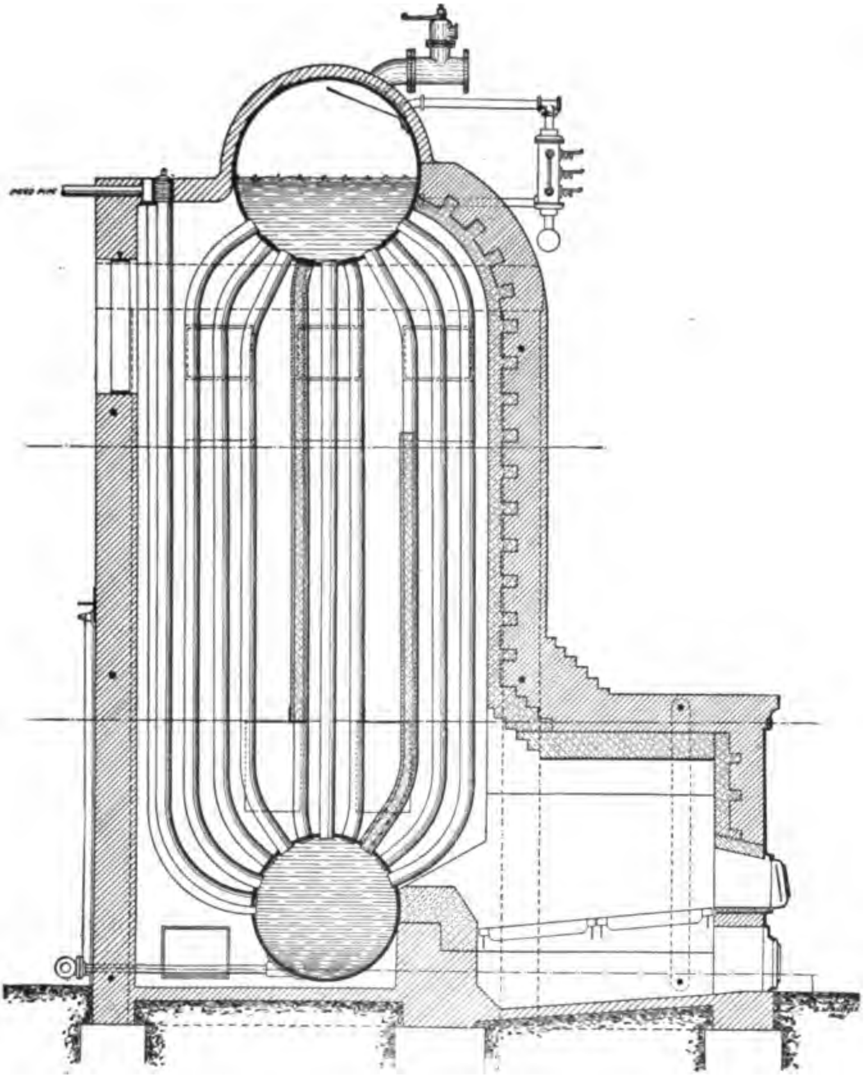


Fig. 65.

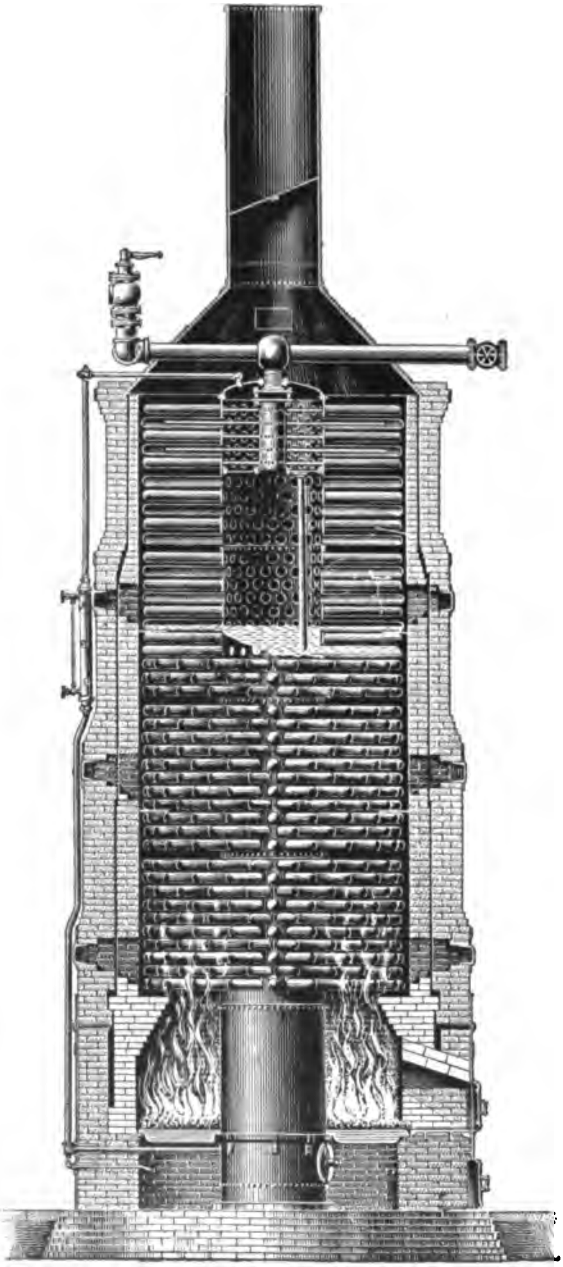
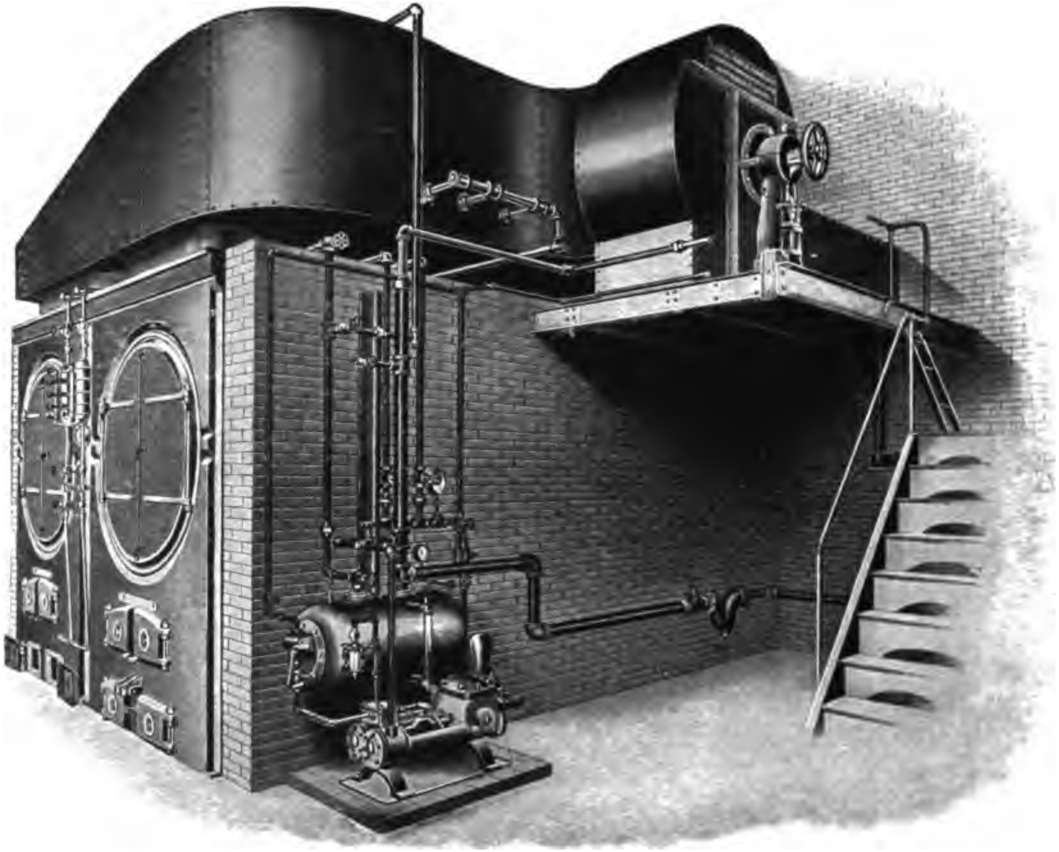


Fig. 66.



STEAM FAN FOR INDUCED DRAFT
At the Works of the Buffalo Forge Company.

the ends. The hot gases are guided by division plates or tile so that they traverse 65 feet of tube-heating surface before they enter the flue. The tubes, being vertical, do not become covered with fine ash, nor do they become clogged with sediment and scale.

Circulation. The feed water enters the row of tubes at the extreme left and flows downward to the mud drum. It then rises as it becomes heated in the hotest generating tubes and enters the steam drum as steam and water. This method of feeding keeps the cold feed water out of the steam drum, and as the cold tubes containing the feed are placed in the path of the escaping gases, but little heat escapes to the chimney.

PECULIAR FORMS.

HAZELTON OR PORCUPINE.

Water Tubes Horizontal—Steam and Water Drum Vertical—Straight-Tube—Single-Tube.

The Hazelton water-tube boiler differs in many ways from the boilers thus far described. Like most water-tube boilers it consists of a steam and water drum and water tubes, but the central standpipe is vertical and the short horizontal tubes radiate from the central drum. According to our classification it is not a vertical water-tube boiler because the tubes are horizontal, also, it is not a horizontal boiler as in general appearance it is vertical.

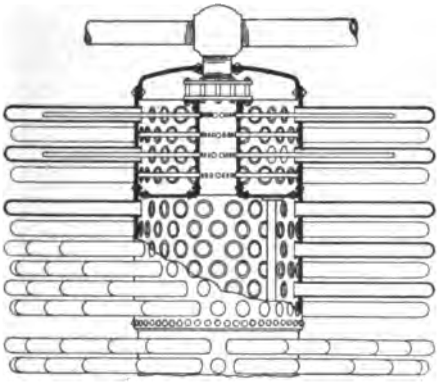


Fig. 67.

The grate is circular and formed around the central drum which rests on a circular cast-iron foundation. Above the grate, the central drum forms part of the heating surface and is the steam reservoir; below the grate it is the mud drum, which may be entered by means of a manhole just below the grate. As shown in Fig. 66, the standpipe above the fire is provided with radial tubes. The appearance of these tubes gives the name "porcupine".

The standpipe is about three feet in diameter for large boilers. The tubes are about four inches in diameter, and two and one-half feet long, the number varying with the capacity of the boiler. The outer ends of the tubes are closed and hemispherical, and the inner ends expanded into the standpipe. These tubes are free to expand and contract without bringing any strain on the boiler.

Steam is taken from the top of the central drum. To get dry steam, small pipes are inserted as shown in Fig. 67. The steam passes up into the small tube at the top of the standpipe and then through the small pipes to the ends of the generating tubes. It then flows back through the generating tubes to the annular space and from thence to the steam pipe. The feed pipe enters the mud drum and extends upward nearly to the water line; it then returns nearly to the level of the grate, terminating in a spraying nozzle.

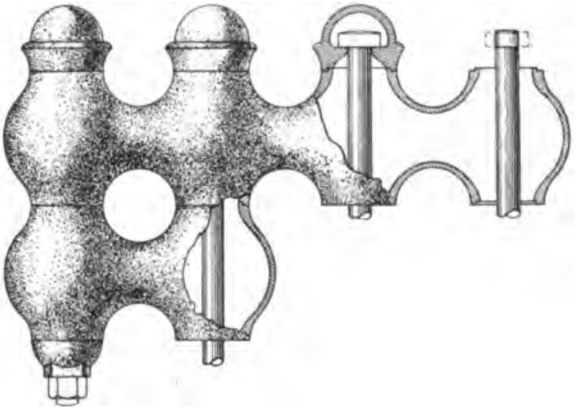


Fig. 68.

This type of boiler may be enclosed in a brick setting as shown in Fig. 66 or by a sheet steel covering lined with fire brick.

HARRISON.

Sectional—Hollow Cast-Iron Spheres Instead of Tubes.

All boilers thus far described have employed tubes as a means of dividing the water into small masses in order to make the heating surfaces more effective. In the Harrison Safety Boiler (Fig. 69) tubes are not used; instead, the water is contained in hollow cast-iron spheres, called units. These units, see Fig. 68, are arranged in vertical rows, called slabs, which are suspended side by side, about one inch apart, from an iron framework. The brickwork setting is merely a covering to keep the hot gases in contact with the units; it does not support the boiler, and can be repaired without disturbing the units.

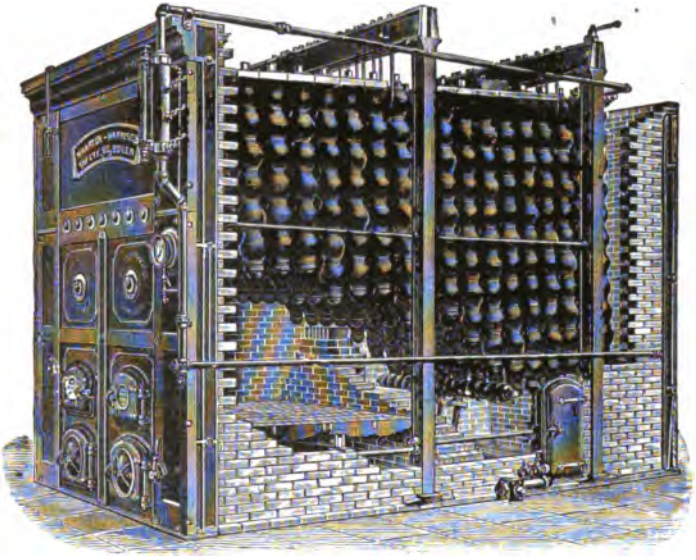
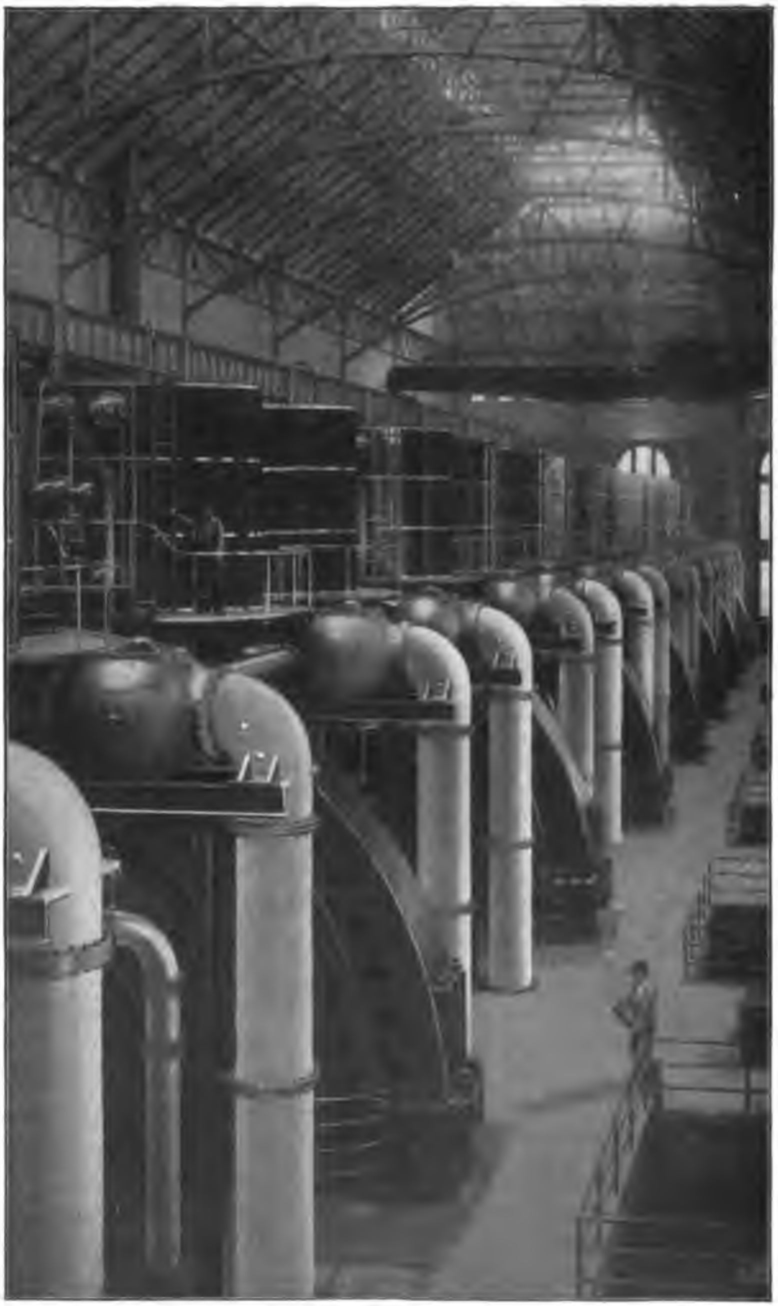


Fig. 69.

The use of units in place of tubes combines great strength and a large heating surface. They are strong because small and spherical and on account of the division of the water into small masses, the heating surface is effective. The units are held together by long bolts which pass through the centers as shown in Fig. 68. The machined faces make a steam-tight joint without packing. This boiler requires the same fittings as other boilers.

The great advantage of this boiler is safety. From the construction, it is apparent that rupture cannot extend beyond the unit; thus disastrous explosions cannot occur. They are claimed to be durable, economical, rapid steamers, and easily handled. The capacity can be increased by merely adding more slabs.



MANHATTAN 74TH ST. POWER STATION, NEW YORK.
Showing Carey's Magnesia Pipe and Boiler Covering.

BOILER ACCESSORIES

PART I

BOILER SETTING

The *setting* for a stationary boiler consists of the foundation and as much of the furnace and flues as is external to the boiler shell. Some internally-fired boilers—the "Lancashire," for instance—have flues in the brick setting. The whole furnace and sometimes the flues, as is the case with the plain cylindrical boiler, are in the setting. Vertical boilers have simply a foundation; and locomotive boilers have no setting, since they are supported by the frames of the engines. Marine boilers are usually placed on saddles, which are built into the framing of the vessel.

In setting a boiler, there are three principal requisites that should be kept in mind: 1. A stable support or foundation for the shell, so arranged as to allow for proper expansion of the boiler. 2. Properly arranged spaces for both furnace flues and ash-pit. 3. A covering which will prevent loss of heat by radiation, and which will not allow moisture to accumulate in contact with the plates.

There are two principal methods for support—by *brackets* riveted to the shell plates, and by *suspension from overhead girders* by means of hooks, rings, etc. In any case the supports should be so arranged that each shall bear its proper proportion of the load and at the same time allow for expansion. If the boiler is short, brackets are generally used; while for long, plain cylindrical boilers the girder method is the more common. If a very long, cylindrical boiler is supported only at each end, the great weight between the two supports is likely to cause bending and an excessive strain on the middle plates, tension in the bottom plates, and compression in the top plates.

The first requisite for a setting is a good *foundation.* If the ground is firm and favorable to a solid foundation, the excavation need be only three or four feet below the level. If it is soft, the excavation should be deeper, and the extra depth filled in with broken stone mixed in with cement, gravel, etc.; or, for very heavy work, piles may be driven. The first course of the foundation should be large stones laid in cement; upon this stonework the walls may be built, either of stone or brick, to within about six inches of the floor-level; and above this, brick should be used.

Sometimes the bed is made of concrete about two feet in thickness. If the soil is very firm, a foundation of large stonework about three feet wide may be built under the side, middle, and end walls only.

In determining the *area* of the bed, the weight that is to be put on each square foot should be estimated carefully. With ordinary condition of the soil, this should not exceed 2,000 pounds. For greater weights, special construction must be used.

The *supporting* and *enclosing* walls are built upon the foundation, with the outer walls at the sides and rear double, the space between, usually about two inches, being an air-space insulation to prevent loss of heat. Projecting bricks, which extend from the outer until they just touch the inner wall, allow for expansion without decreasing the strength of the inner wall. The side walls are strengthened by buckstays or binders, which are kept in place by long bolts, secured by nuts on each end. Fig. 1 shows a boiler in the brick setting, supported by brackets, the front brackets resting on iron plates which are built into the walls; the rear brackets, being supported by rollers, are free to move as the shell expands. If designed for anthracite coal, the distance between the shell and the grate-bars is about two feet; for softer coal, this distance is increased a few inches.

The furnace is lined with firebrick, both front and sides; and sometimes portions back of the bridge, as well as the bridge itself, may thus be protected. The space between the bridge and the shell is from 6 to 8 inches, which brings the hot gases into close contact with the boiler before they enter the combustion chamber beyond, the rear and side walls being built a little higher than the top row of tubes. The fire-line must not be carried above the water-line; if it is, the intense heat is likely to injure the shell-plates. Never expose any part of the boiler not covered by water to the flames from the furnace. The side walls are built about the same height as the rear walls. The space at the rear is bridged over and stiffened by T-irons. In order to increase the heating surface, the top is arched so that the hot gases will pass over the steam space before they enter the chimney.

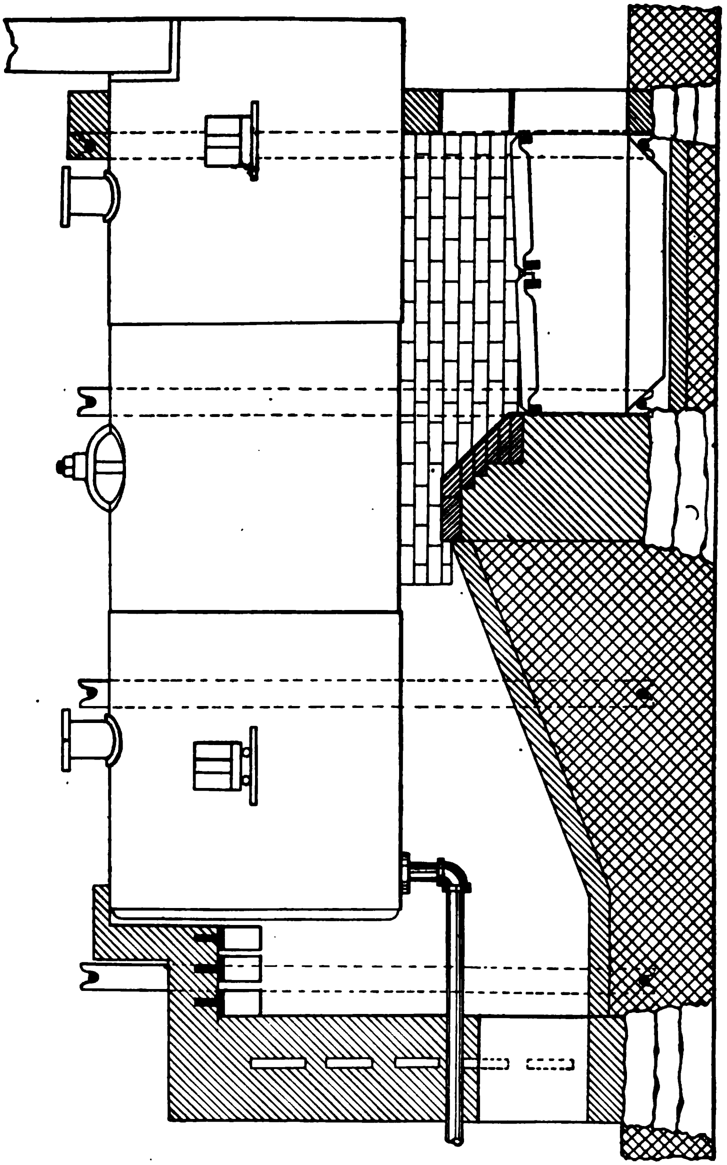


Fig. 1. Side Elevation of Boiler in Brick Setting.

The smoke box projects over the front end of the boiler and has a rectangular uptake.

Fig. 3 shows the top view of the same boiler.

The front is usually of cast iron, with doors for firing and cleaning and for access to the tubes. Soot, dirt, etc., are removed through the door in the brickwork at the rear.

The end which contains the handhole should be set about one inch *lower* than the other end, so that the sediment and detached scale will tend to accumulate there.

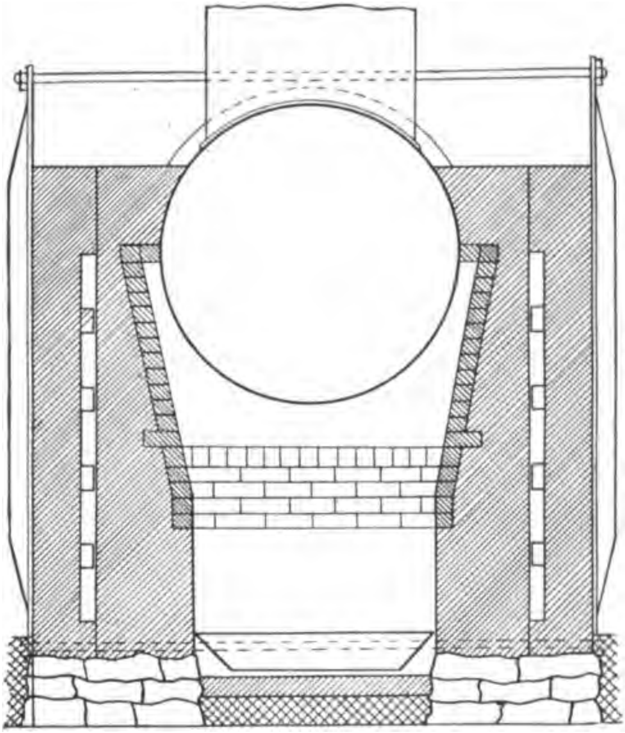


Fig. 2. Front Elevation of Boiler in Setting, Showing Binders Bolted in Place, to Strengthen Side Walls.

Internally-fired boilers may also be enclosed in brickwork. The setting is a support and covering, forming the side flues but not the furnace. Excess of brickwork surface in contact with the shell, should be avoided, as brickwork collects moisture, which causes external corrosion.

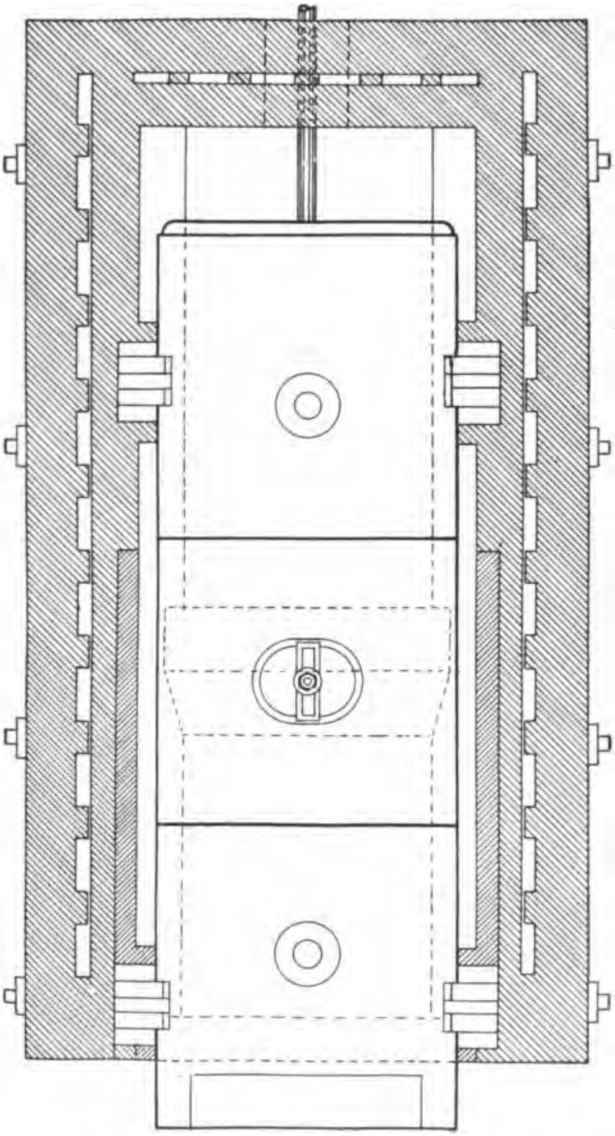


Fig. 8. Plan of Boiler and Setting.

Water-Tube Boilers. The settings for water-tube boilers are similar to the settings of cylindrical tubular boilers. Marine water-tube boilers are enclosed in sheet-iron casing, which is lined with non-conducting material, usually asbestos or magnesia.

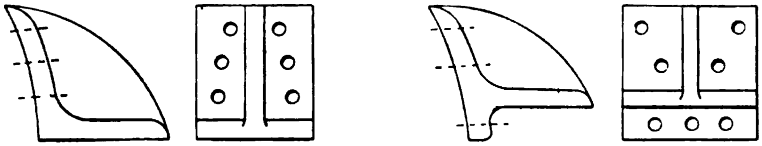


Fig. 4. Fig. 5.

Types of Brackets for Supporting Boilers.
In Fig. 4 the rivets are all above the flange; in Fig. 5 they are both above and below the flange.

Supports. There are, as already intimated, two common methods of supporting boilers—1. By means of *brackets*; 2. By *suspending from wrought-iron beams.*

If the boiler is about 15 feet long, it is customary to use two brackets on each side. If more than 15 feet, three on each side are used. The front brackets rest on the brickwork, but the others rest on small iron rollers to allow for expansion. Brackets are so arranged that the plane of support will be a little above the middle. There are several forms of brackets. The form shown in Fig. 4 is usually made of cast iron, and is provided with rivets above the flange of the bracket. It is better to have the rivets both above and below the flange, as shown in Fig. 5.

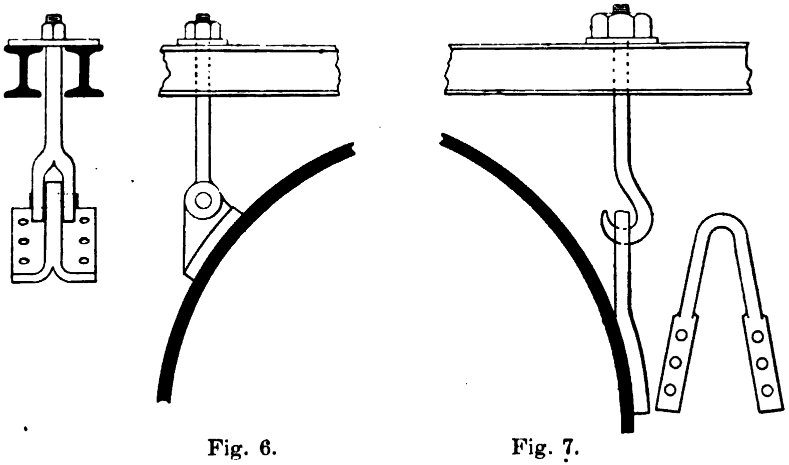


Fig. 6. Fig. 7.

Two Methods of Supporting Boilers by Suspending from Overhead Beams.

Fig. 6 shows one method of suspending from beams. A lug, made of wrought iron, is riveted to the plates of the boiler. A bolt having one end bent like a hook, holds the lug from the beam. In Fig. 7 the lug is replaced by a loop of wrought iron. Fig. 8 shows another method of suspension, the connection between the rod and the boiler-plates being short pieces of boiler-plate arranged for flexibility.

When the boiler is of small diameter, it may be suspended as shown in Fig. 9.

FURNACES

To get the maximum efficiency from any boiler, it is necessary that the fuel shall be properly consumed, and that the proportions of the furnace shall be such as to give the maximum results. No boiler is economical the furnace of which is so small that the fire has to be forced to obtain the desired result. The furnace, of course, will vary in shape, size, and detail with the type of boiler and the kind of fuel; but certain essentials—such as doors, grate-bars, bridge, and ash-pit—are similar in all furnaces. To obtain the maximum effi-

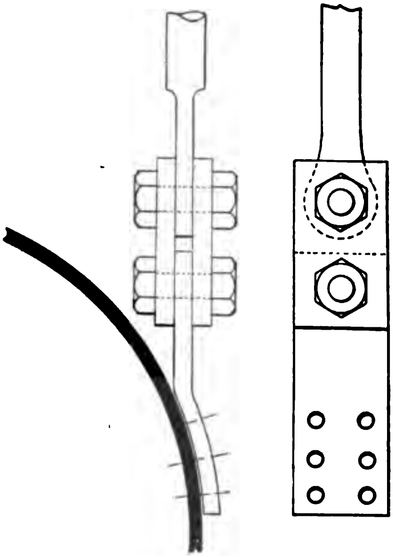


Fig. 8. Flexible Support for Suspended Boiler. Flexibility Secured by means of Two Pieces of Boiler-Plate Bolted Together.

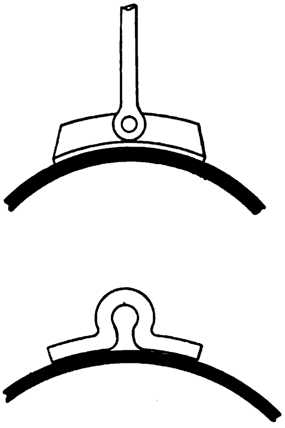


Fig. 9. Illustrating Method of Suspending Boiler of Small Diameter.

ciency of combustion, there should be a uniform and abundant supply of air to the under side of the grate. This is easily obtained when the boilers are externally fired, but may be somewhat restricted when they are internally fired. If smoky fuels are used, a moderate supply of air is necessary on the surface of the coal, to prevent excessive smoke formation; but, as the air thus admitted is usually cold, the quantity should be small, to prevent unnecessary cooling of the furnace. This air is generally supplied through a draft-plate in the fire-door.

All possible radiation should, of course, be prevented. In the case of internally-fired boilers, this radiation is not likely to be excessive, for most of the heat would have to pass through the water in the boiler before radiating, and it is a comparatively easy matter to encase such a boiler in some sort of approved lagging which will prevent most of the heat from escaping. The case is somewhat different with the externally-fired boiler, where the furnace is built in a mass of brickwork below the boiler. In such a furnace a considerable amount of heat may radiate directly from the fire without coming in contact with the boiler or water at all.

To allow for complete combustion, there should be a sufficient space between the grate and the boiler. In externally-fired boilers, this space may be approximately two feet. If this distance is increased beyond proper limits, some effect of the heat will be lost; and if the distance is small, the plates are likely to be damaged, and complete combustion impaired. In the internally-fired boiler, the combustion space is frequently sacrificed in order to obtain a large grate area. If the space between the grate-bars and the boiler is too small to allow complete combustion, a combustion chamber must be provided immediately back of the bridge, which will permit of the complete combustion of the gases. The ideal place, of course, for the combustion chamber, is immediately over the grate. In locomotive boilers, the crown sheet is usually four to six feet above the grate; but such a height is manifestly impossible in marine or other internally-fired boilers, and the combustion chamber behind the bridge wall, in the Scotch boiler, partially compensates for the loss of space immediately over the grate.

The incandescent fuel and unconsumed gases should not come in contact with the cold surfaces of the boiler if the most efficient combustion is desired. This condition is violated in internally-fired boilers, where the fire comes directly against metal having water on one side of it. If the flame is chilled by contact with cold surfaces before the gases are completely burned, a considerable amount of smoke is likely to result.

The fire-grate should be of such dimensions that the fireman can work efficiently. A grate more than six feet long cannot be properly taken care of at the farther end; and if the grate is more than four feet wide, two fire-doors should be provided. The height of the grate should be laid out with proper reference to the floor, two feet above the floor being about right. If the grate is high, it is difficult, if not impossible, to tend the fire properly. These conditions are dependent, not so much upon the boiler, as upon the physical limitations of the fireman, and of course are eliminated by using the mechanical stoker.

To the above conditions may be added a suitable temperature in the fire-room. No man can tend a fire properly in excessive heat. In stationary work it is not difficult to maintain proper conditions in the fire-room; but at sea, where the supply of air is necessarily limited to what can come in through small openings, it is a different problem. The fire space on board ship is small; and the air coming through the ventilating ducts usually makes an exceedingly cold spot immediately under the duct without producing much effect in other parts of the room.

Door. The furnace door is usually made of cast iron, and is supplied with a circular or sliding draft-plate or *grid*, which admits air to the top of the fire as needed. It is usually protected by a perforated, wrought-iron baffle-plate bolted to the door casting inside, with an air-space of two or three inches between. This not only protects the cast iron of the door from the direct force of the flame, but it forms a chamber for the proper distribution of the air-supply, and also helps to heat it somewhat before reaching the furnace.

In many of the French torpedo-boats, a patent swinging door is provided, set on horizontal hinges swinging inwards. The door, of course, must be held open while the stoker is tending the fire; but in case a tube blows out, it prevents the rapid escape of steam into the fire-room. This is a matter of much more importance in the restricted fire-room commonly found on a vessel than it would be on land.

Grate. The size of grate will depend upon the quantity of coal likely to be burned. For ordinary draft, this may be 15 lbs. or upward per square foot of grate surface per hour; for forced draft, 40 to 60 lbs.; and in some cases as much as 100 lbs. per square foot of grate surface has been burned. If the grates are long, they are usually inclined slightly downwards, say ¾ inch to the foot, which is a great assistance in firing and makes it easier to keep fire on the farther end of the grate.* The grate-bars are usually made of cast iron, as this material is cheaper than wrought iron and in most instances lasts as well. The bars are made in various forms, according to the fuel burned and the shape of the firebox.

For large grates, the bars are made singly or in pairs. For smaller grates, they are made in larger groups. Grate-bars should not be more than three feet in length. The length of grate can easily be a multiple of the length of these bars. The bars have distance pieces at the ends, and perhaps in the middle, to prevent distortion. They are usually 3 inches or more in depth at the middle, tapering to perhaps an inch or so at the ends; and the cross-section is slightly tapered from top to bottom, so that the bars can easily be withdrawn from the sand after casting. They are usually made a trifle shorter than the place in which they fit, to allow for expansion, 2 per cent of the length of the bar usually being sufficient for this purpose. The air-spaces between the bars are usually about ½ inch in width. For burning pea coal or screenings, a finer grate must be used. For anthracite coal, the space may be a little larger. Bituminous coal, which readily cakes, can have a considerable space between the bars—and this, indeed, is essential for a proper supply of air.

Fig. 10 shows a circular grate, such as is placed in a vertical boiler. *M* shows the style of grate-bar used in burning sawdust or shavings; *N* is what is known as the *herring bone* grate; and *O* is a group of bars of the ordinary form. In locomotives, and in boilers where the grates are subjected to extra hard usage, wrought-iron bars may be used. The point of fusion of wrought iron being higher than that of cast iron, the former would possess a considerable advantage were it not for the fact that wrought iron will bend and twist more readily than cast iron. Grates have been made of hollow bars, through which water is caused to circulate. By this method their durability is increased, and the *water-grate* forms a fairly good feed-water heater. This type of grate, however, is expensive.

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* The grates have an incline of a few inches, so that the bed of coal will be thicker at the rear than at the front; this allows a more even consumption of fuel, as the air passes through the fire at the bridge more freely.

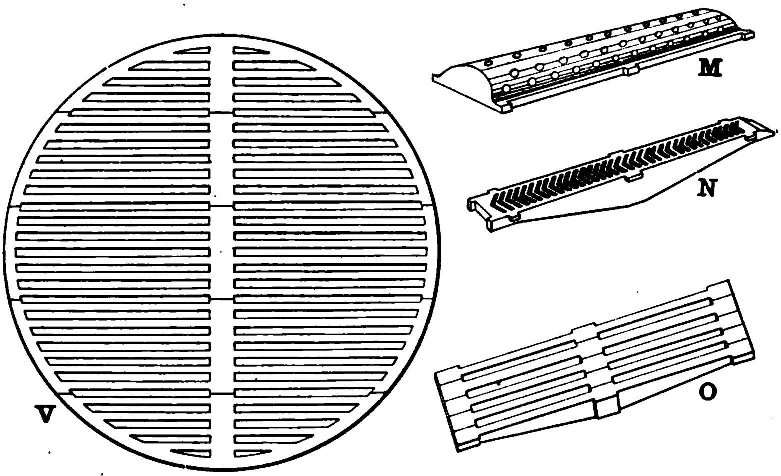

Fig. 10. Types of Grates for Boilers. *V*—Circular Grate for Vertical Boiler; *M*—Grate for Burning Sawdust or Shavings; *N*—"Herring-Bone" Grate. *O* Group of Grate-Bars of Ordinary Form.

Rocking Grates. The labor of breaking the clinkers is considerable when ordinary fixed grate-bars are used; and to economize this labor, various forms of rocking-grates have been devised. In locomotives, rocking-grates are essential; and since the rate of combustion is high, the fire must always be kept in good condition; and

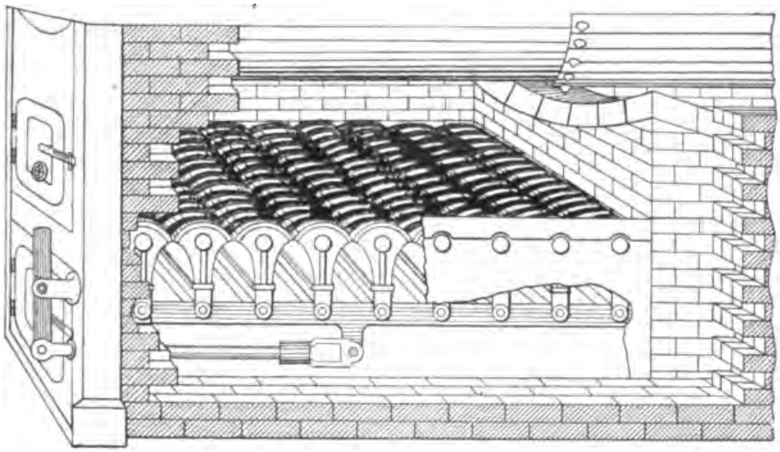

Fig. 11. "Kelley Standard" Rocking Grate.

the grate, being below the cab floor, cannot easily be reached by hand. Fig. 11 shows the "Kelley Standard" rocking grate. Each bar is made up of a number of separate leaves, which can be removed and replaced without renewing the whole bar. When the bar is moved back and forth by means of a lever outside the brickwork, the leaves oscillate through a small angle and break up the clinkers.

Another form of bar, shown in Fig. 12, has proved very satisfactory. *A* and *B* are two bars, the ends of which are of different depths. These rest at each end on a crank-shaft *C*. As this is oscillated by the lever *G*, the alternate bars move up and down, and the clinkers are easily shaken out.

Bridge. The bridge is a large wall or partition at the back of the grate, usually built of firebrick or cast iron, or of ordinary brick covered with firebrick. The bridge separates the grate from the combustion chamber, and causes the gases to come in close contact with the boiler in passing into the combustion chamber. The proper height of the bridge will depend upon the draft. If the space is narrow between the bridge wall and the boiler, more draft will be necessary to carry the gases through. Two or more bridges may sometimes be built in long boilers to keep the gases in contact with the shell as long as possible.

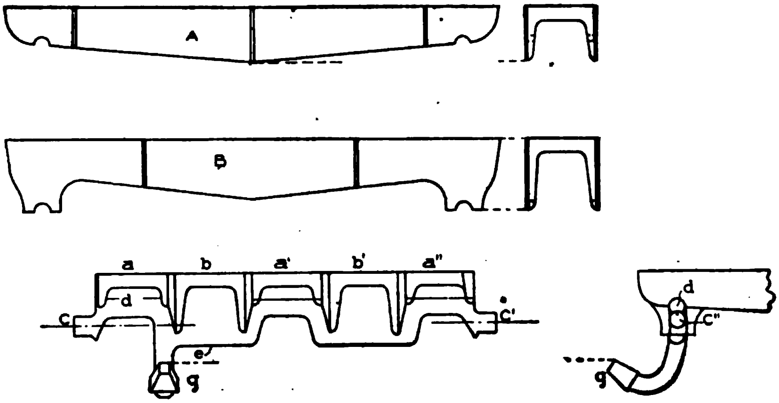

Fig. 12. Rocking Grate Consisting of Alternate Bars with Ends of Different Depths Resting on a Crank-Shaft Oscillated by a Lever.

Special Furnaces. Almost any furnace is adapted for the use of anthracite or bituminous coal containing less than 20 per cent of volatile matter; but if there is more than this amount of volatile matter, the heat is likely to be so intense that the fire should not be

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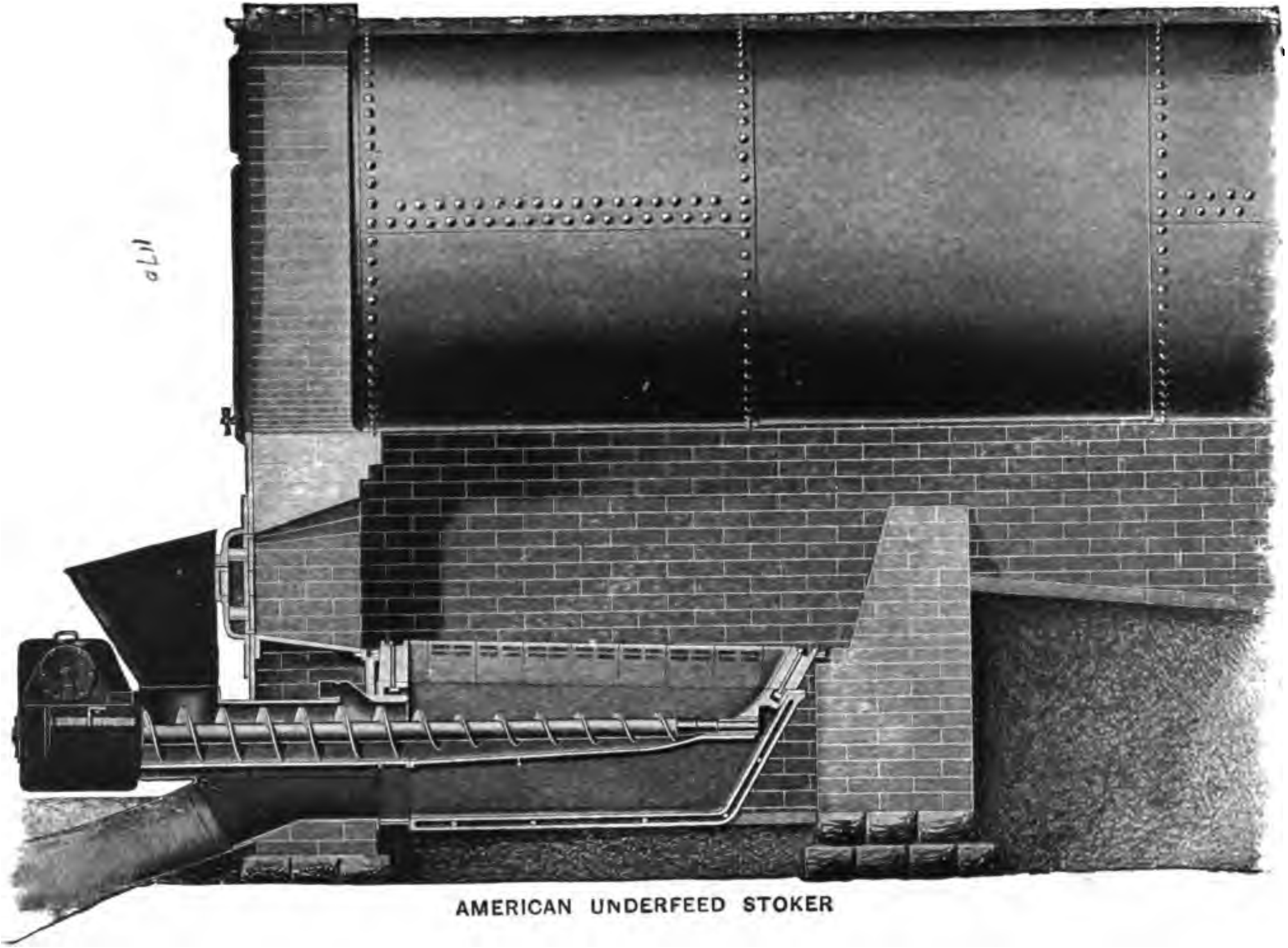


AMERICAN UNDERFEED STOKER

brought in direct contact with the boiler. If the fuel should contain 40 per cent of volatile matter, the furnace should be surrounded with firebrick and should have a high combustion chamber. Coal is the most common fuel used; but wood, sawdust, and straw are not uncommon fuels. When these are burned, there should be plenty of room in the furnace, and a sufficient supply of air on top of the fuel. Sawdust, shavings, and fine coal may be blown into the furnace by an air-blast.

In the West, crude petroleum is becoming a common fuel. Experiments have shown that one pound of crude oil is equivalent in heat units to something less than two pounds of good coal. Oil has many advantages as a boiler fuel. It is clean, gives a uniform heat, is economical, and requires much less attention than coal. There are no ashes to handle, and one man can easily tend two or three times the number of furnaces that he could if burning coal. The fire can be started and stopped instantly; and the supply of air can be so regulated that, unless the boiler is forced to the limit, there will be practically no production of smoke. Whether or not oil is an economical fuel, will depend upon the local conditions and the market.

Oil fuel is fed into the furnace through a sprayer formed, in some cases of two concentric conical tubes. Compressed air or steam entering through the one tube draws the oil through the other, on the principle of the atomizer, and throws it into the furnace in a fine spray. For marine work, compressed air should be used, as the loss of steam for this purpose would be a matter of considerable consequence. Steam, however, is sometimes used in marine work, in which case the vessel must be equipped with an evaporator to make up the steam thus lost. On land, where fresh water is plenty, steam is usually preferred, and is less expensive in first cost.

Prevention of Smoke. In large cities, where the escape of considerable quantities of smoke is undesirable, several methods have been devised either to consume the smoke or to prevent its formation. The cause of smoke, as we have seen, is an insufficiency in the supply of air, or perhaps a too abundant supply of cold air above the fire; or, again, smoke may be due to the contact of the flame with cold surfaces. An exceedingly high temperature is necessary to consume the finely divided particles of carbon, and anything that tends to chill the flame will cause smoke.

The actual loss caused by the escape of smoke, even when it is dense and black, has been found to be slight, and usually the appliance used for prevention costs more than is saved. The alternate firing of two furnaces which open into a common combustion chamber, or the alternate firing of two sides of the same furnace, produces a slight gain if the proper amount of air is admitted. But if, in order to burn the smoke, the bed in one furnace or on one side of a furnace is allowed to become thin, there will be no gain in efficiency.

The introduction of steam is an efficient method, but it is likely to cause a too rapid rate of combustion.

Another arrangement to prevent the escape of smoke is that by which the coal is distilled in a small furnace which is separate from the boiler. The coke and gases thus made are burned in the furnace of the steam boiler. This device is not altogether satisfactory, on account of the loss of heat from the detached furnace. Rather than add any smoke-prevention device, anthracite or coke may be used instead of bituminous coal.

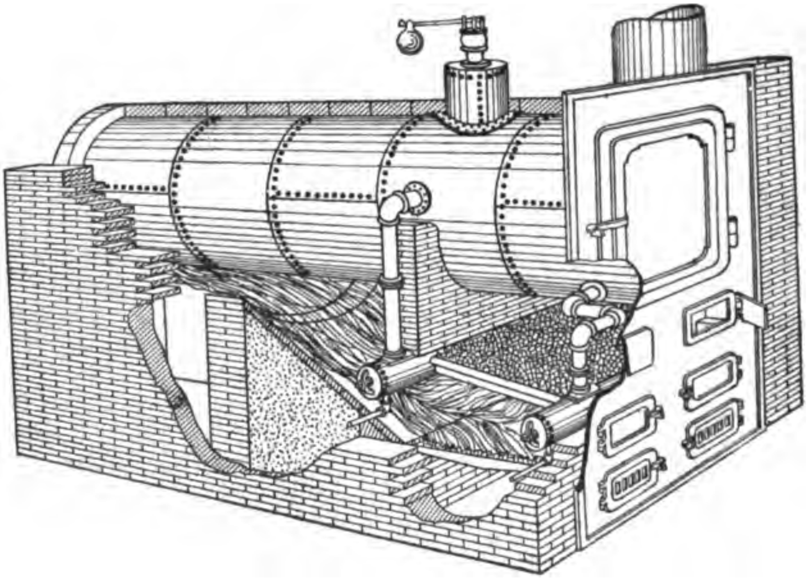


Fig. 13. "Hawley" Down-Draft Furnace Attached to Horizontal Multitubular Boiler. Note Upper Grate Consisting of Water Tubes Connected to Steel Drums.

Many engineers and business men consider *a good fireman* to be the best smoke preventer.

Down-Draft Furnaces. In order to increase economy and capacity, or to prevent smoke, a down-draft furnace is sometimes used. In this type of furnace, there are two grates, one a foot or more above the other. Fresh coal is fed to the upper grate, and, as it becomes partially consumed, falls through to the grate below, where the combustion is completed. The draft is *downward* through the upper grate, and *upward* through the lower, because the connection to the chimney is from the space between the grates. The volatile gases are carried down through the bed on the upper grate, and are burned in the space below it, where they meet the hot air drawn upward from the lower grate. A large proportion of the air for combustion enters the door at the upper grate. Tests on the Hawley furnace show that 30 to 45 pounds of coal per square foot per hour can be burned with good results.

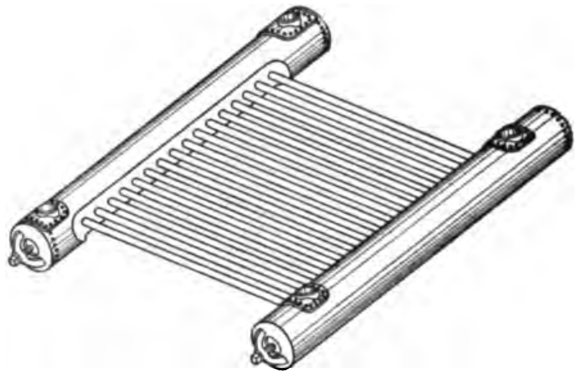


Fig. 14. Upper Grate of "Hawley" Down-Draft Furnace. The Grate-Bars are Water Tubes Connected to Steel Drums which are Connected to Boiler.

In the furnace made by the Hawley Down-Draft Boiler Company, the grates are formed of a series of water tubes opening at the ends into steel drums, shown in Figs. 13 and 14, which are connected with the boiler. Fig. 13 shows this furnace attached to a horizontal, multitubular boiler. It may be applied to both *tubular* and *water-tube* boilers with good results, and is advantageous to boilers of insufficient heating surface, and when inferior fuels are burned. It is claimed that this attachment insures complete combustion, small amount of ashes on account of the second grate, good water circulation, and increased economy and capacity.

The Hollow Arch. Among boiler accessories specially adapted for use on locomotives because of their intense draft, the hollow arch has recently come into prominence. Its principle is simply that of a conduit providing a passage for the admission of heated air to the firebox above the fire, in addition to the air that comes up through the grate from below in the ordinary way. Its object is to keep the supply of oxygen at all times sufficient in quantity, and at the proper temperature, to insure a practically perfect combustion of the unconsumed carbon and hydrocarbon gases which are ordinarily wasted and lost in the form of black smoke pouring from the stack. It thus insures an economy of fuel and a proportional reduction in operating expense.

The problem of securing complete combustion of fuel on a locomotive, is one that presents peculiar difficulties. The quantity of fuel to be burned is so large, and the firing space relatively so small, that the conditions usually are unfavorable for economical combustion. A ton of average bituminous coal contains about 1,000 pounds of pure carbon, 700 pounds of hydrocarbon gases, and 300 pounds of noncombustible matter or ash. The 1,700 pounds of carbon and hydrocarbons require about 300,000 cubic feet of air for their complete combustion. In the ordinary method of burning coal on a locomotive, fully 90 per cent of this air—or 270,000 cubic feet per ton of fuel burned—must be drawn up through the grate-bars and firebed. This is practically impossible without forcing the draft to such an extent that the fire will be pulled off the grates, and more or less of the unburned coal carried away through the flues and stack. The result is that the supply of air actually used is, as a general thing, insufficient for perfect combustion, and the combustible carbon smoke and gases pass out of the stack without giving up all of their heat to the water in the boiler. The energy they contain is simply wasted.

How, then, can this be prevented? In other words, since the quantity of air that comes through the grates is insufficient, how can we get enough air to the fuel without interfering with the fire? It must be let in *above the fire*; but it will not do to admit cold air, which, as every fireman knows, would act as a damper on the fire, retarding combustion, and increasing rather than preventing smoke and loss of energy. The air to be admitted to the fire must first be *heated to as near the ignition point as possible.*

This is done by means of the *hollow arch.* One of these arches of the "Wade-Nicholson" type, installed on a locomotive, is illustrated in Fig. 15, the method of operation being clearly indicated. The device may be installed at both back and front ends of the firebox.

The hollow passage through the arch leads directly through suitable openings in the firebox sheets, from the outer air to the combustion chamber, being deflected downward toward the fire at the inner end. The walls of the arch, being highly heated, impart their heat to the current of air, which, as it emerges into the firebox, is practically at the temperature of ignition. There mingling directly with the combustible gases, an approximately perfect combustion is established. The resulting economy in fuel is estimated to average a saving of at least 8 per cent.

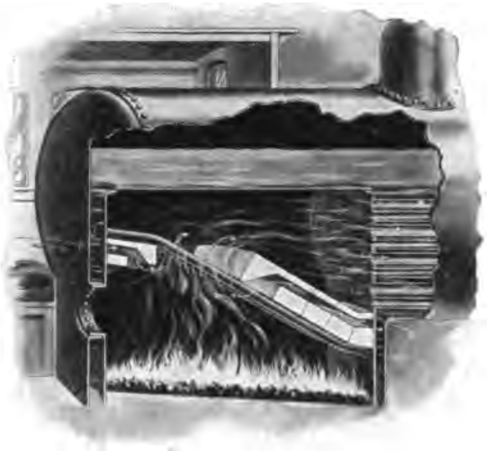


Fig. 15. Wade-Nicholson Hollow Arch Installed in Locomotive Boiler. The Water-Tube Supports Here Shown are Sometimes Omitted.

The Chicago & Northwestern Railway, has, after severe test, adopted arches of the above type on over 200 of its locomotives; and its example has been followed on many of the locomotives of the Santa Fé, the Chicago, Milwaukee & St. Paul, the Père Marquette, the Duluth & Iron Range, and other important railroads in this country. In addition to the saving in fuel, the following advantages are claimed for the hollow arch:

Being air-cooled, its life is two to three times that of the ordinary solid brick arch.

It does away with the smoke nuisance.

The air, being heated before striking the combustible gases, unites with them instantly, giving a brighter, cleaner, more intense fire, and resulting in a better steaming engine.

The back arch acts as a baffle-sheet, protecting the crown sheet and upper flues, and gives a more uniform distribution of heat throughout, resulting in less leaky flues and a saving in boiler repairs.

The arch can be used either with or without water-filled circulating arch tubes as supports.

Arches can readily be removed and reset, in whole or in part, without damage, to give access to flues when repairs are needed.

Fuel Economizers. Many devices have been employed whereby a portion of the heat may be extracted from the gases as they pass

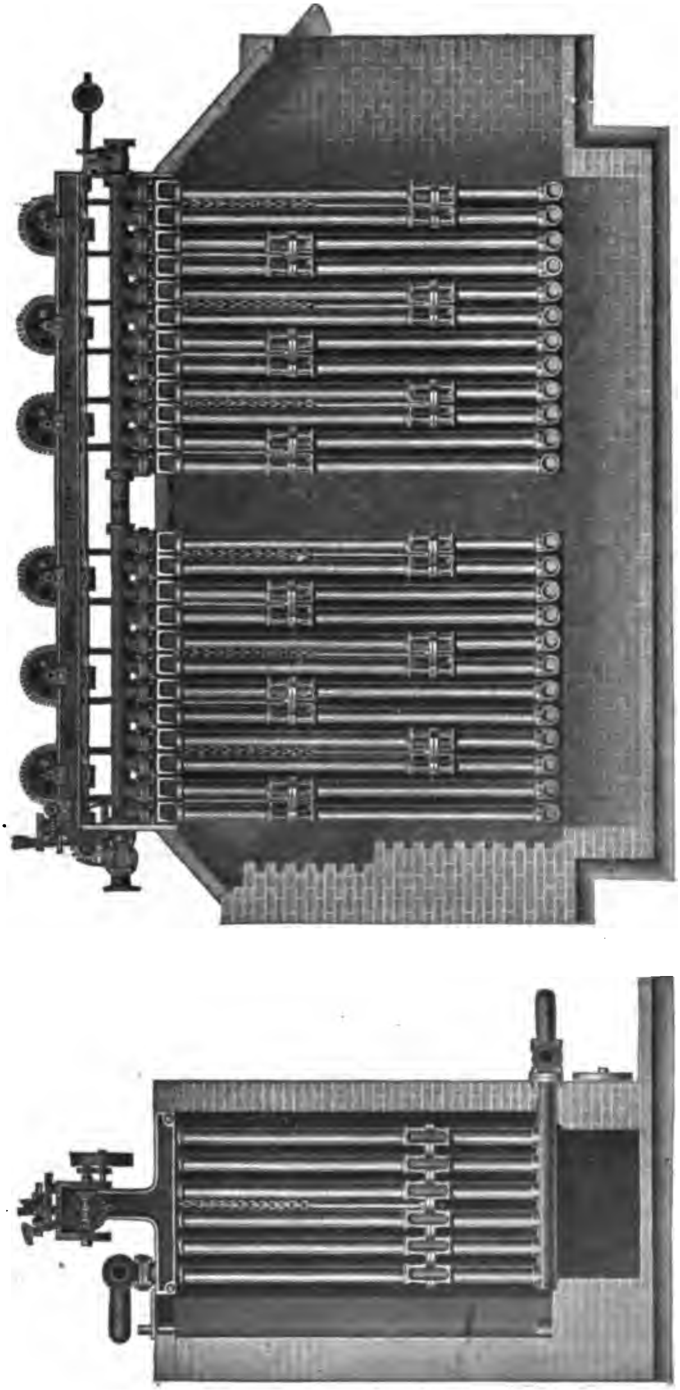


Fig. 16. Green Economizer.

from the boiler to the uptake. Most of these consist of a tubular arrangement through which the hot gases pass; but, as these are soon covered with a thick deposit of soot, they quickly become inoperative. The "Green" economizer (Fig. 16) solves this difficulty by means of small scrapers which work up and down between the tubes. These scrapers are operated by a small engine, and keep the tubes free from soot. The feed-water is pumped through these tubes on its way to the boiler, and is thoroughly heated. An economizer of this sort will extract 40 per cent or more of the heat from the waste gases; but by reducing the temperature of these gases, the draft is somewhat reduced, and either the chimney must be built higher, or a blower must be used.

Mechanical Stokers. The mechanical stoker, which feeds coal and tends fires by machinery, is coming more and more into general use. With a good mechanical stoker, one man can tend several furnaces with little labor. There are several different types, and in most of them the coal is fed into a hopper of such size that it need not be often filled. Some stokers work continuously; others, only when thrown into gear by the fireman. In the "Roney" stoker (Fig. 17), the grate-bars extend across the furnace, and form a series of steps down which the fuel moves. Each grate bar is hung on pivots at the ends, and is operated by a rocker-bar. This rocker-bar is driven by a small steam engine, with a slow, regular reciprocation which causes the grate-bars to tip so that the coal of its own weight slides from one grate-bar to the next. Coal from a hopper falls onto a horizontal plate, and is fed into the top of the grate by a *pusher*. The rapidity with which the fuel can be fed, is regulated by changing the stroke of the pusher and by governing the speed of the engine. Ashes

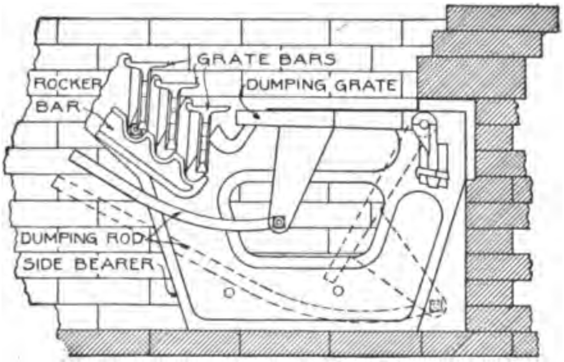


Fig. 17. Detail of "Roney" Mechanical Stoker.

and clinkers collect on the dumping-grate at the end of the grate-bars, whence they can be dumped into the ash-pit.

This type of grate is well adapted for smoke prevention, for the fresh fuel fed in at the top is rapidly coked, and the volatile gases are easily consumed. The rapidity of feed should be so regulated that no unburned fuel gets past the dump-grate. If the fire becomes too thin, there will be a loss of efficiency due to the excess of air which passes through the burning fuel. It is easy to detect the loss from too much fuel, but not so easy if there is too little fuel.

All mechanical stokers in which the movable parts are inside the furnaces, are likely to get out of order because of the heat and dirt.

Fusible Plugs. Fusible plugs are usually inserted in the top sheet or crown sheet of boilers, as a safeguard against collapse of the furnace crown should the water in any way be drawn out of the boiler while the fire is burning. These plugs consist of a core composed of an alloy of tin, lead, and bismuth, with a covering of brass or cast iron. The United States inspection law requires at least one fusible plug to be put in every marine boiler, with the exception of water-tube boilers, the plug to be made of a bronze casing filled with good-quality "Banca" tin from end to end. While this plug is kept at a comparatively low temperature by water on one side, the fire on the other side will not melt it; but when the water-level becomes low enough to leave one end of the plug uncovered, the alloy core of the plug, having a comparatively low melting point, will fuse, thus running out of its casing, relieving the pressure in the boiler, and allowing the excess of steam to extinguish the fire, which otherwise would be likely to destroy the crown sheet.

Fusible plugs are frequently unreliable. Sometimes they will blow out when there is no apparent cause, and sometimes remain intact when the plates have become overheated. If a coating of hard scale is allowed to accumulate over the plug, it may stand considerable pressure, even after the core has become melted. To provide against this, the plug should be replaced frequently. If allowed to remain in the boiler for any length of time, the composition of the alloy is likely to change, the plug thus becoming more or less unreliable.

Figs. 18 and 19 illustrate the ordinary plug. It should be so made that, when screwed into the crown sheet, it will project 1½ or

2 inches above the plates, so that when the alloy melts there will be a sufficient depth of water over the crown sheet to prevent injury from heat.

Sometimes the core is covered with a thin copper cap, as shown in Fig. 18, which protects the alloy from contact with the water, thus preventing a chemical change and the formation of scale. It does not necessarily follow that a hole ½ inch or ¾ inch in diameter will liberate steam fast enough to prevent excess of pressure. If a small quantity of steam is introduced into the firebox, it may have the

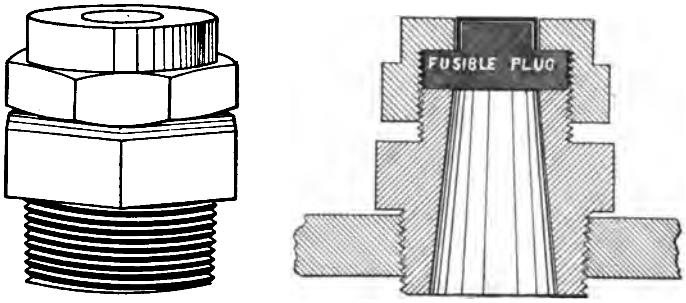


Fig. 18. Fusible Plug. At Right is Sectional View of Plug Attached to Crown Sheet of Boiler, to Give Automatic Warning in Case of Overheating of Plates.

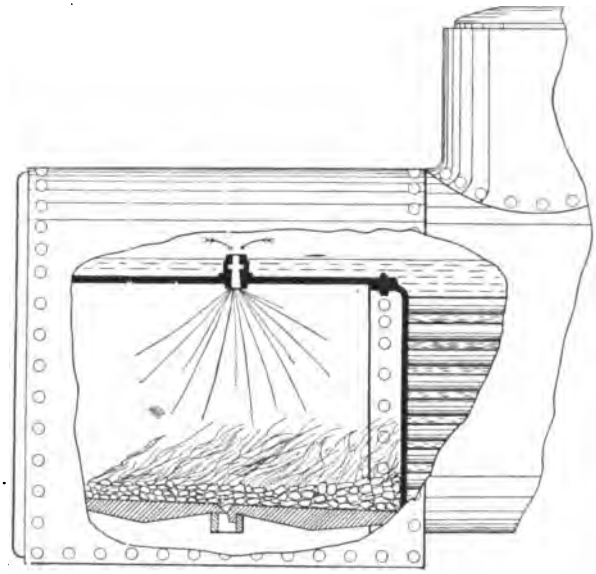


Fig. 19. Illustrating Action of Fusible Plug Attached to Crown Sheet.

effect of brightening the fire and increasing the heat of combustion, owing to the formation of water gas as the steam mingles with the burning coal. The steam, moreover, might have the effect of inducing additional draft. If, however, the quantity of escaping steam and water is considerable, combustion will be retarded, and the fire will be partially extinguished. It will operate to warn the fireman of what has happened; and if the escape of steam is not too rapid, he may throw on wet ashes and deaden the fire.

NATURAL AND FORCED DRAFTS

The draft in a chimney is caused by the difference in weight between the volume of heated gases inside and the outside air. This being so, it is apparent that the taller the chimney, the greater this difference will be. The force or intensity of a draft is increased, and additional draft is induced, by the force of the wind as it whistles by the chimney top. The intensity may at any time be measured by a *draft-gauge.* The most satisfactory instrument of this sort is the "Eames Differential" draft-gauge, shown in Fig. 20. The tube is filled with a special non-drying, non-evaporating oil of known specific gravity. The incline and diameter of the tube are so proportioned that the readings are equivalent to inches of water, in which terms the draft is invariably measured.

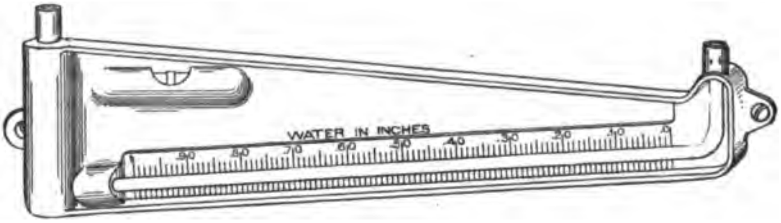


Fig. 20. "Eames Differential" Draft-Gauge.

Other things being equal, the rate of combustion depends upon the height of the chimney. A chimney 20 to 25 feet in height will cause a draft sufficient to burn about 8 lbs. of coal per square foot of grate area per hour. If the height is increased to about 100 feet, the rate of combustion will be increased to approximately 15 lbs. per square foot; and to burn 25 lbs., the chimney should be about 175 feet high. This is measured above the grate of the boiler. For good bituminous or anthracite coal, the chimney must be higher than for wood, if the same rate of combustion is desired. If the boiler has small or winding passages, the chimney must be higher to produce the same effective draft. High chimneys are costly; and it is frequently the practice to build two or three small chimneys in place of the big one, and to supplement them with some form of *forced draft.*

By means of forced draft, the rate of fuel combustion can be increased under favorable conditions to 100 lbs. of coal per square foot of grate surface per hour. This, of course, greatly increases the power of the plant, but is likely to injure the boiler, and is uneconomical under most conditions. There are three systems of forced draft in common use:

1. The *closed stoke-hold,* as used in marine work;
2. The *closed ash-pit;*
3. The *induced draft.*

Closed Stoke-Hold. One of the most common forms of forced draft, especially as used on warships, is obtained by closing the stoke-holds and blowing a fresh supply of air into the fire-room. This gives an exceedingly good ventilation and keeps the fire-room in good condition; but its chief objection is that when the furnace doors are opened there is a tremendous indraft of cold air, which tends to lower the efficiency of the boiler. If this system is employed, the bulkheads adjacent to the boiler-room must be provided with double doors, forming an air-lock between. By opening only one door at a time, the pressure in the fire-room is not lost. This system seems to possess but one distinct advantage, and that is coolness and therefore comfort for the firemen; but the disadvantage of the inrush of air to the furnaces when firing, is sufficient, in some cases, to make the system questionable.

Closed Ash-Pit. The essential features of forced draft by this method consist merely in closing the ash-pit tight, and blowing the air directly under the grate. When the fires are cleaned, the draft, of course, must be shut off; otherwise the flames will be blown out into the fire-room. The fire-room, under this system, is likely to be hotter than by the other method; but this system would seem to be the better from a mechanical point of view.

There are several patented devices in connection with the forced draft, of which the "Howden" and the "Ellis and Eaves" systems may be specially mentioned. It may be worth while to note that if fuel-oil is burned, any one of these systems of forced draft will work better than with coal, for the fire can be tended without opening the fire-doors.

Induced Draft. Perhaps the most common example of induced draft is to be found in the locomotive, where the exhaust steam is turned into the smokestack. The rush of this steam up the stack, by carrying a large volume of air with it, induces a tremendous draft. Induced draft may also be obtained in stationary and marine plants by placing a blower in the chimney or stack. In marine work, of course, induced draft by exhaust steam is out of the question. When a blower is placed in the smokestack, an economizer should be used, so that the gases may be cooled before they reach the blower. The draft obtained on locomotives is frequently equivalent to a column of five or six inches of water; while a forced draft of two inches is usually considered large, except for torpedo-boats, which may have as strong a draft as a locomotive has.

Howden System. The Howden system of forced draft with closed ash-pit has been used to a considerable extent in both mercantile and naval service. The air supplied to the ash-pit is first heated by passing through a heater in the uptake. Waste gases pass through tubes; and the air, passing among them before entering the furnace, is heated to a high temperature. A consumption of 60 lbs. of coal per square foot of grate is easily obtained with this system; and care must be taken that the fire is not forced too hard, as there is more danger of burning out the grate than if the air-supply is not heated.

Ellis and Eaves System. Heating the air does not necessitate its being forced into the closed ash-pit, for it is quite feasible to heat the air in connection with draft induced by an exhaust fan at the base of the funnel. Such is the Ellis and Eaves system. This system was first tried in the boiler shops at the works of the John Brown Company, in Sheffield, England, and was later adopted on many vessels. The Ellis and Eaves heater is fixed on top of the boilers, and is divided into two parts separated at the front by a smoke-box and at the back by a funnel. The hot gases, therefore—which pass outside the tubes—have to take a somewhat circuitous course; while the passage of the air to be heated, on the contrary, takes a direct course. The distribution of air to the ash-pit is similar to that of the Howden system.

The advantages of this system lie in the general convenience of the induced draft and the absence of jets of hot air shooting out into the boiler-room. The draft need not be shut off when stoking the fires, unless it is desired to prevent the inrush of air already referred to under the general discussion of "closed stoke-holds." The air in the fire-room being of a relatively higher temperature than would obtain with closed stoke-holds, and the quantity being much less, this objection has no great weight. With the Howden system it is necessary that the doors should be tight; otherwise hot air will be blown out into the fire-room. With this system a few leaks are of no consequence, and the fire-room will be somewhat cooler than with the Howden System. The objections to the Ellis and Eaves system are those inherent in any system of draft induced by a fan—that is to say, a poor efficiency of the fan working in heated gases, and lost work in drawing air through tortuous passages.

Steam Jets. Steam jets may be used for inducing a draft. They may be placed either in the smokestack, or below or above the grate; but in general they are not so economical as a fan used for the same purpose. In locomotives and fire-engines, where the exhaust steam is at high pressure, an intense draft may be induced by exhausting this up the smokestack. In both these cases, the saving of weight due to the use of a small boiler running at high tension, is of greater practical importance than the economy of fuel; and for such purposes this arrangement is entirely satisfactory.

A steam jet may be used directly in the furnace, either above or below the grate. The steam enters through a small pipe, and expands through a nozzle surrounded by an annular, funnel-shaped tube. The escape of steam from the inner nozzle draws in a large volume of air through the outer tube, and produces an intense draft. If steam is blown into the ash-pit in this manner, it forms a sort of producer gas by mingling with the incandescent fuel, and materially aids in the combustion of cheap and apparently worthless fuel. Almost as poor fuel can be successfully used with this arrangement as can be used in the grates of the down-draft furnaces. Such arrangements have given excellent satisfaction, and the production of smoke is materially lessened.

Some tests made in the French Navy some years ago, showed that, with the use of the steam jet above the grate, the coal consumption per square foot of grate area could readily be doubled; but this result would be attained at the expense of fuel economy; for, while with natural draft one pound of coal produced approximately eight pounds of steam which could be used by the engine, with a steam jet less than 6$\frac{3}{4}$ pounds of steam per pound of coal was available for like purposes. The total evaporation per pound of fuel was approximately the same in each case, the difference being the quantity of steam used in the jet. If a steam jet is used on board ship, it consumes a considerable amount of fresh water, which must be replaced by evaporators, or by the use of salt water, which is decidedly objectionable.

TUBE-CLEANERS

To secure the best results from a boiler, the tubes should be kept thoroughly clean. The collection of soot on the tubes is as detrimental to economy as the formation of boiler scale. The soot may be removed by the insertion of brushes when the boiler is not under steam, or the tubes may be blown out with a steam jet designed for this purpose. Fig. 21 illustrates forms of tube-cleaners, of which there are numerous types on the market. The type shown at *B* is designed for use with a steam jet. In the case of oil-burning locomotives, the tubes are usually cleaned with the aid of a sand-blast.

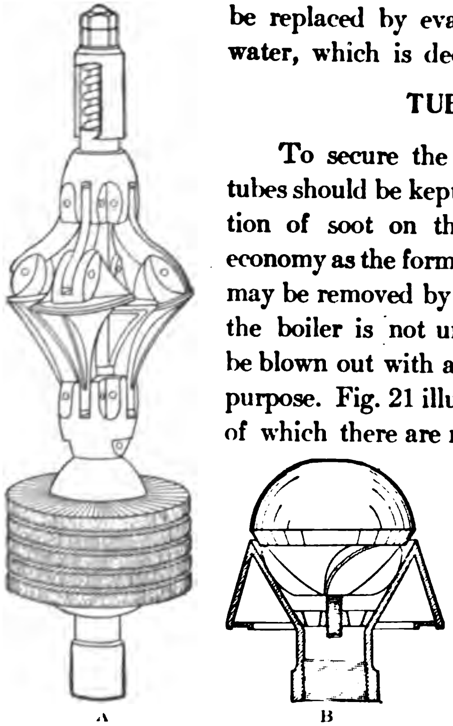


Fig. 21. Types of Tube-Cleaners.

TUBE-STOPPERS

It frequently happens, when tubular boilers are under pressure, that leaks occur in the tubes through pitting, defective welding, or the development of cracks. Formerly, when this occurred, the fire was drawn, and the ends of the tube plugged with hardwood bungs driven hard home or with iron plugs calked in. With high pressures, such procedure is impossible. Tube-stoppers used for high pressure are joined together by a tie-bar of some sort. They are usually wedge-shaped; and the tie-rod, passing through the stopper at one end, with a plug at the other end, can be screwed hard up.

The simplest form of stopper has to be inserted from the rear, and necessitates drawing the fire; but Fig. 22 illustrates a stopper which can be inserted without drawing the fire. At the end of the rod is hinged a folding bung, which can be passed through the tube and which opens out in the combustion chamber before being pulled into position. At the smoke-box end of the boiler, an india-rubber washer, pressed between two pieces of metal, affords temporary protection while the plug is being put in position. The stopper can then be screwed up tightly with a handle provided for that purpose.

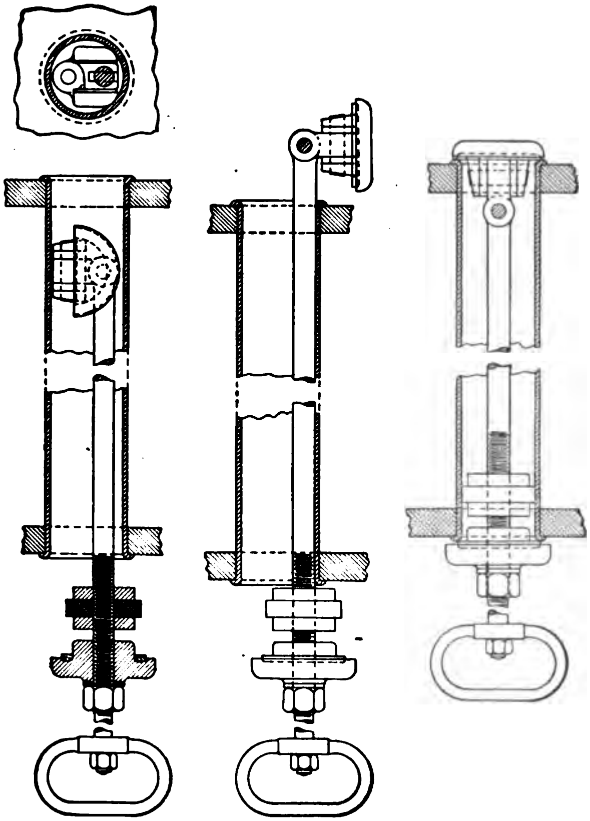


Fig. 22. Tube-Stopper Designed for Insertion without Drawing Fire.

Fig. 23 illustrates another arrangement which can be inserted in the leaky tube without drawing the fire. The ends, being in the form of stuffing glands, press an asbestos packing hard against the side of the tube.

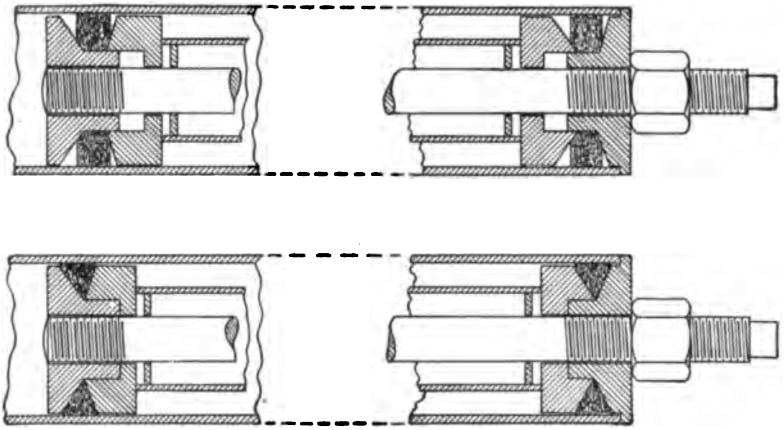


Fig. 23. Another Type of Tube-Stopper Used without Drawing Fire. As the Parts are Screwed Up, the Asbestos Packing is Driven Hard against Side of Tube.

MANHOLES AND HANDHOLES

A *manhole* allows access to the boiler for cleaning and repairs. It is usually elliptical in form and large enough to admit a man. About 16 inches for the major axis, and 12 for the minor axis, is a good size. The manhole is closed by a plate or cover made of cast or wrought iron. This plate is held to the seat by a yoke or yokes, and bolts. Fig. 24 shows one form, *Y* being the yoke, *L* the cover, and *N* the bolt. The joint between the cover and the shell is made steam tight by packing.

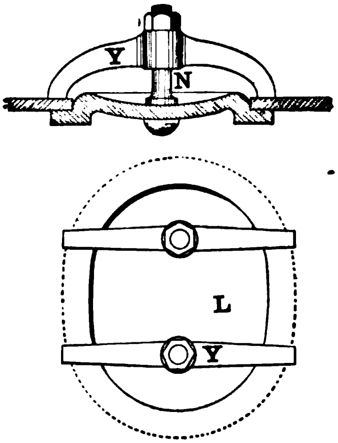
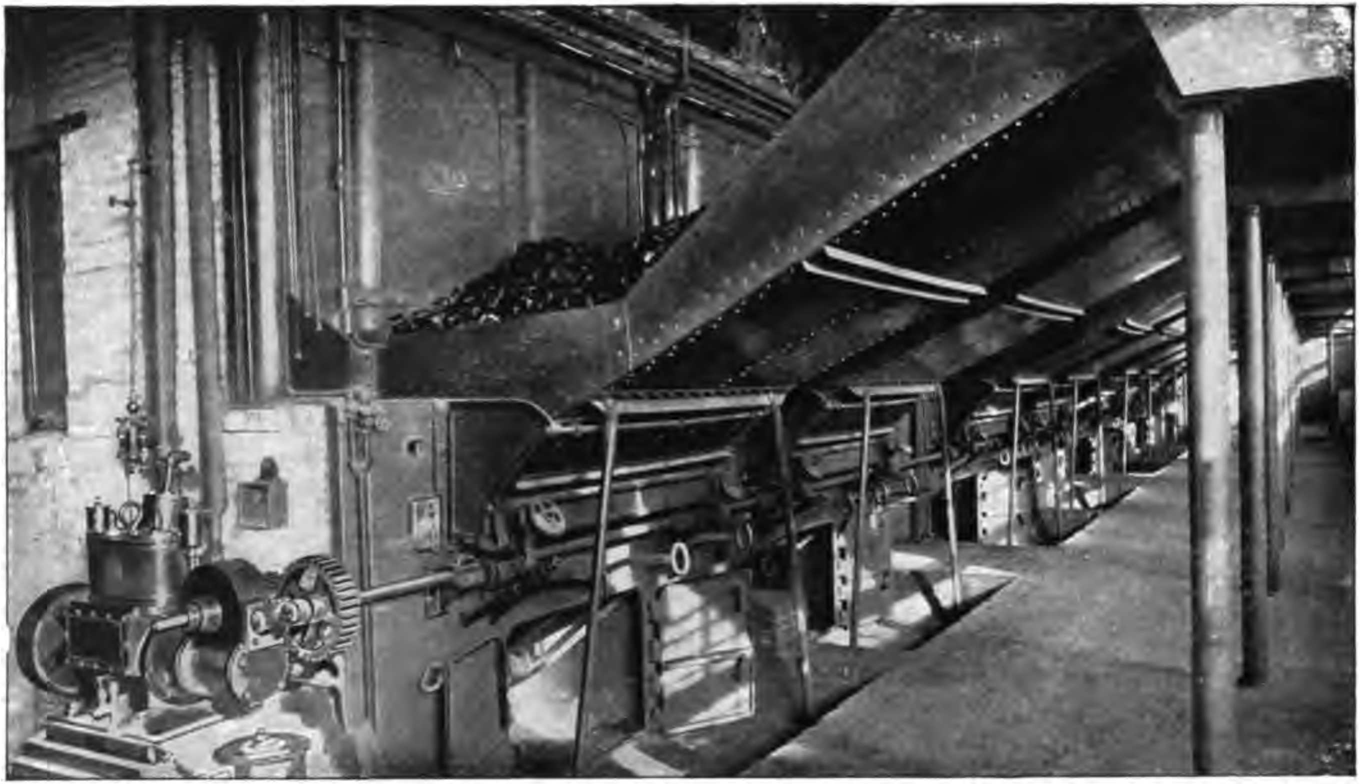


Fig. 24. Elevation and Plan of Manhole Cover.

The *strength* of the boiler should always remain unimpaired; so, whenever a large hole is cut in the plate, the edge should be strengthened, for the tension is concentrated there, and the plates are, moreover, likely to become weak by corrosion. The strain put upon the plate by screwing up the cover, if no packing is used, is considerable, especially if a piece of scale gets between the faces and the joint is then made tight.



Roney Mechanical Stoker, Showing Coal Delivery Shutes and Operating Engine.

Fig. 25 shows the section of a strong and simple manhole. The edge of the plate is strengthened by a broad ring of steel, which is flanged and riveted to the shell, its edge forming the seat. The cover as shown in the figure is shaped for strength. The edge of the ring which forms the seat, and the cover, are machined to make a tight joint without packing. The strengthening ring should be at least $\frac{5}{8}$ inch thick and 4 inches wide, that the rivet-holes may not be too near the edge.

Handholes and *mudholes* are more commonly placed in boilers, which are so constructed that a man cannot enter—in a vertical boiler, for example. They are used to some extent in other boilers; in horizontal return-tube boilers there is usually a handhole in each end, near the bottom. Handholes are very convenient to admit hose for washing out the boiler, also for removing scale and sediment. Handholes are similar to manholes in construction, but require only one yoke and one bolt to keep them in place. Mudholes should be provided in order that the sediment and detached scale can be removed without lifting the accumulated mass to the top manhole. Mudholes and handholes greatly facilitate cleaning the fire-box water-leg of locomotive and small vertical boilers.

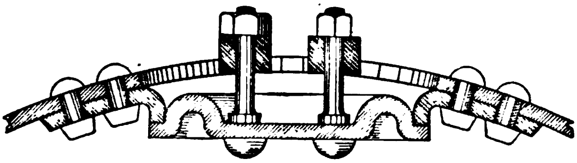


Fig. 25. Section of a Strong but Simple Type of Manhole.

STEAM AND VACUUM GAUGES

The steam pressure in the boiler is measured in *pounds per square inch.* When we say the boiler is working or steaming at 80 pounds' pressure, we mean that the gauge pressure is 80 pounds; that is, the pressure in the boiler is 80 pounds above *atmospheric pressure.* It could be measured by a water or mercury column; but, as these would need to be very high to measure the pressures used at the present day, they are not practicable, and so a spring-pressure gauge is used instead.

The dial gauge, now used almost universally, was invented by M. Bourdon. It is designed in accordance with the principle that a flattened, curved tube closed at one end tends to become straight when subjected to internal pressure.

The tube, which is usually oval in section, is bent into the arc of a circle as shown in Fig. 26. One end is *fixed*, and is in communication with the boiler. The other is *closed* and free to move. By means of levers, a curved rack, and a pinion, the motion of the free end is *multiplied* and *indicated* by a needle, which is attached to the pinion. The needle moves over a dial which is graduated to agree with a mercury column, or with a standard gauge. The back-lash of the levers is taken up by a hair spring. Fig. 27 shows the interior and face of a Bourdon steam gauge manufactured by the American Steam Gauge Company.

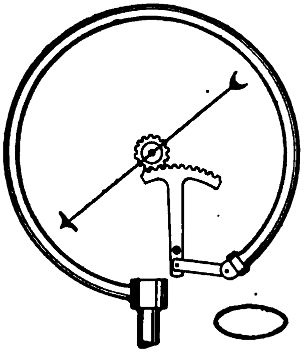


Fig. 26. Steam-Filled Curved Tube Indicating Pressure in Bourdon Steam Gauge.

Fig. 28 shows the exterior and interior of a steam gauge with a light tube for low pressures; the face of the dial is graduated corresponding to the mercury column. The only difference between this gauge and the *vacuum gauge*, is that in the latter the curved tube is turned in the opposite direction so that the needle will move clockwise with a *decrease* of pressure.

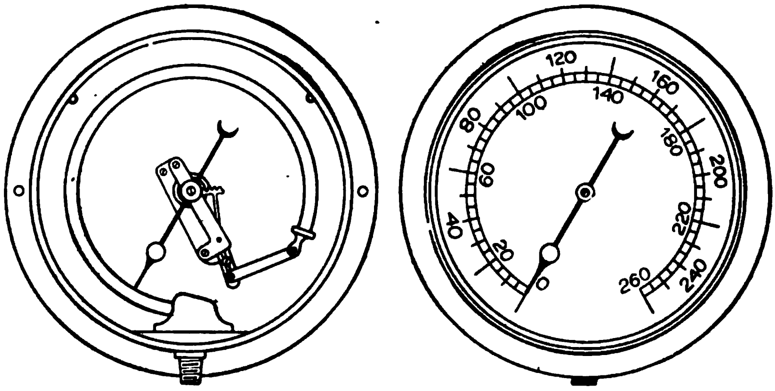


Fig. 27. Interior Mechanism, and Dial, of "Lane" Type of Steam Gauge.

On account of the jarring, the gauge for locomotives must be very strong. To prevent excessive vibration of the needle, two short, stiffer springs are used, as shown in Fig. 29.

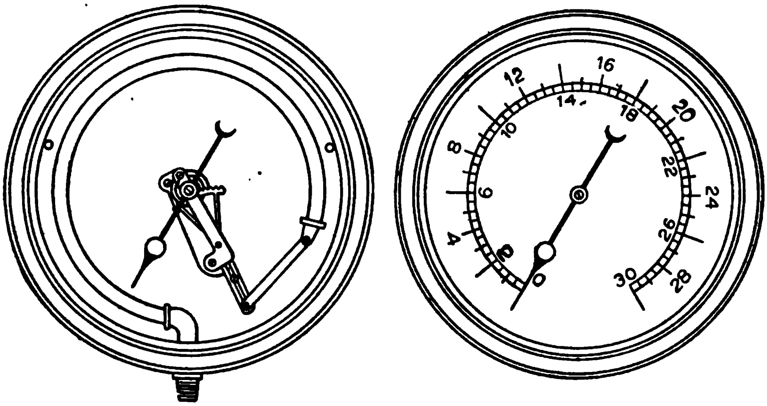


Fig. 28. Interior Mechanism, and Dial, of Low-Pressure Steam Gauge.

Sometimes two pressure gauges are fitted to a boiler, one indicating the working pressure, and the other graduated to about twice the working pressure. The latter is useful in testing the boiler under water pressure, and also serves as a check on the other. The pipe which connects the pressure gauge to the boiler should have bends in it near the gauge. These bends—or, better, a *coil pipe*, as shown in Fig. 30—

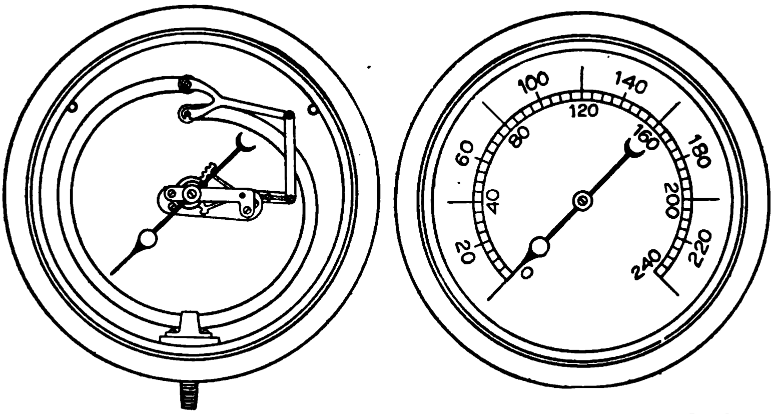


Fig. 29. Steam Gauge for Use on Locomotives. Excessive Vibration of Needle Prevented by Use of Two Short, Stiff Springs.

are filled with water, which transmits pressure and keeps the spring at a nearly constant low temperature. Gauges should be placed where the water in the coiled pipe will not freeze; also, the gauge should not be exposed to strong heat. In order that the gauge may be removed from the boiler for examination, repairs, or calibration, when the boiler is under pressure, the connection should be provided with stop-cocks.

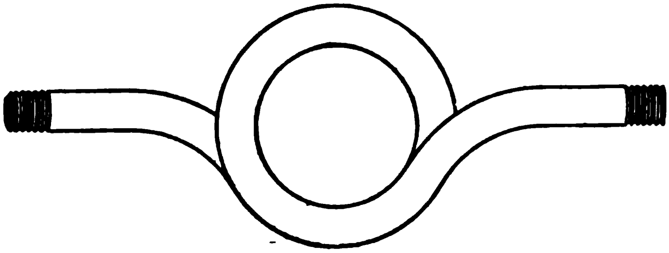


Fig. 30. Water-Filled Coil Pipe for Connection to Steam Gauge. The Water Transmits Pressure and Regulates Temperature.

In a battery of boilers, *each should have its pressure gauge,* which should be connected *directly* to the boiler, *not* to the steam pipe.

WATER GAUGES

It is of great importance that the level of the water in the boiler can easily be ascertained at all times. Should the level be too low, there is danger of overheating the furnace plates or tubes. If it is too high, there is likely to be an undue amount of priming. The water-level is usually indicated by gauge-cocks or try-cocks or water gauge-glasses. Sometimes a float is provided, which is connected to a small whistle, and if the water-level falls below a

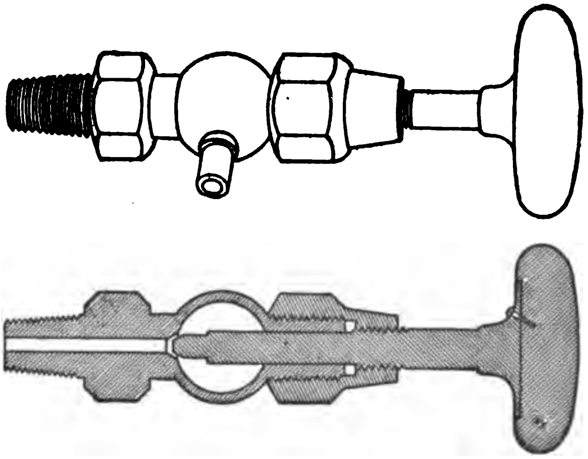


Fig. 31. Ordinary Form of Try-Cock for Determining Water-Level in Boiler.

certain point, an alarm is sounded. Such a device can readily be used in conjunction with the ordinary water-gauge.

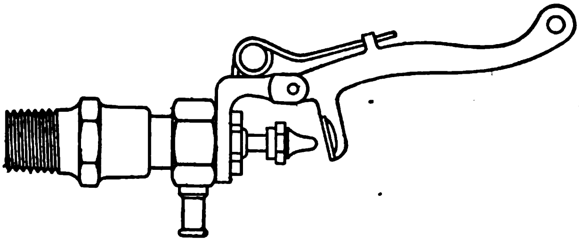


Fig. 32. Try-Cock Operated by Means of Lever.

Try-Cocks. Try-cocks are very generally used. They are of widely different forms, and may be either like the general type shown in Fig. 31, which is the ordinary locomotive form, constructed in two parts so that they can be separated for the purpose of repacking without detachment from the boiler; or they may be of the lever type shown in Fig. 32. There are usually three cocks, one at the highest desired water-level, one at the lowest, and one midway. More cocks may, of course, be used if desired. The water-level can be determined by opening the cocks in succession and observing whether dry steam or hot water flows out. If the boiler is encased in brickwork, as is customary for externally-fired boilers, the gauge-cocks are placed outside the brickwork, and are connected to the boiler by nipples of the proper length.

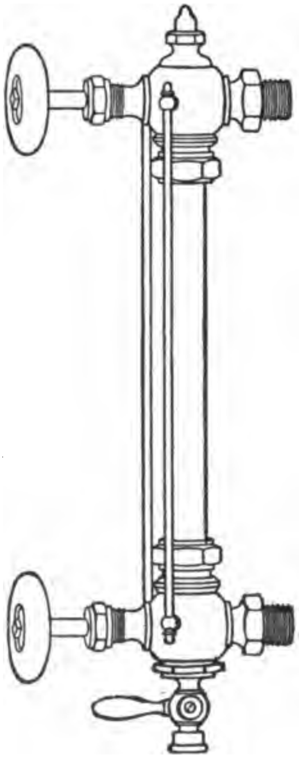


Fig. 33. A Good Type of Water Gauge-Glass.

Gauge-Glasses. In order that the fireman may know the water-level *without* trying the cocks, a water gauge-glass is used. It consists of a strong glass tube about one foot in length, having the ends connected to the boiler by suitable fittings. As both ends of the tube are in communication with the boiler, the water-level in the glass will be the same as in the boiler, and is *always in sight.* Fig. 33 shows a good form of gauge-glass. The glass is protected by rods which are parallel to it. As the glass frequently needs cleaning, repacking, or renewing, cocks are provided for shutting off communication with the boiler. A drain-cock is also placed at the lower end to empty the glass when the attendant wishes to ascertain whether the glass is working properly or not. The drain-cock is often provided with a drain-pipe. The steam and water passages should be at least one half-inch in internal diameter.

The glass is likely to break because of accident or of changes in temperature. To prevent serious injury to the fireman and loss of water as a result of the breaking of the gauge-glass, *automatic valves* may be placed in the passages. In Fig. 34 the ball-valve is shown in detail. If the glass breaks, the pressure of the steam drives the ball outward, filling the conical passage. When a new glass is put in, the balls are forced back by slowly screwing in the stems. This, like other safety devices, is very likely not to work when it should.

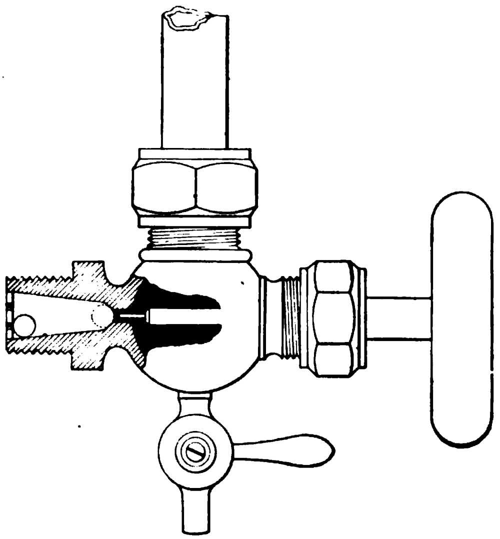


Fig. 34. Automatically Acting Ball-Valve to Prevent Injury to Workmen and Loss of Water on Breaking of Gauge-Glass.

In boilers where the steam space is small, as in locomotives, the allowable variation of water-level is slight; but the greater care with which the glass is watched makes up for the small margin of safety. If dirty water is used, or if the water foams, the level in the glass will be unsteady and unreliable, since dirt clogs the passages, unless they are large, and the foaming causes a fluctuation of the water-level. A small pipe connecting with the steam space where no ebullition occurs, will insure a steadier water-level. If the steam and water connections are long, the pipes should be made large.

The chief objection to the gauge-glass—namely, its breaking—may be to some extent overcome by attaching the gauge-glass to a *gauge-column*, which is usually made of brass and stands quite clear of the boiler itself. In such an arrangement as this, the temperature in the gauge-glass cannot vary so widely as if it were attached directly to the boiler. The "Klinger Patent" water gauge-glass is not easily broken, and possesses many advantages over the common glass. Fig. 35 illustrates both these devices.

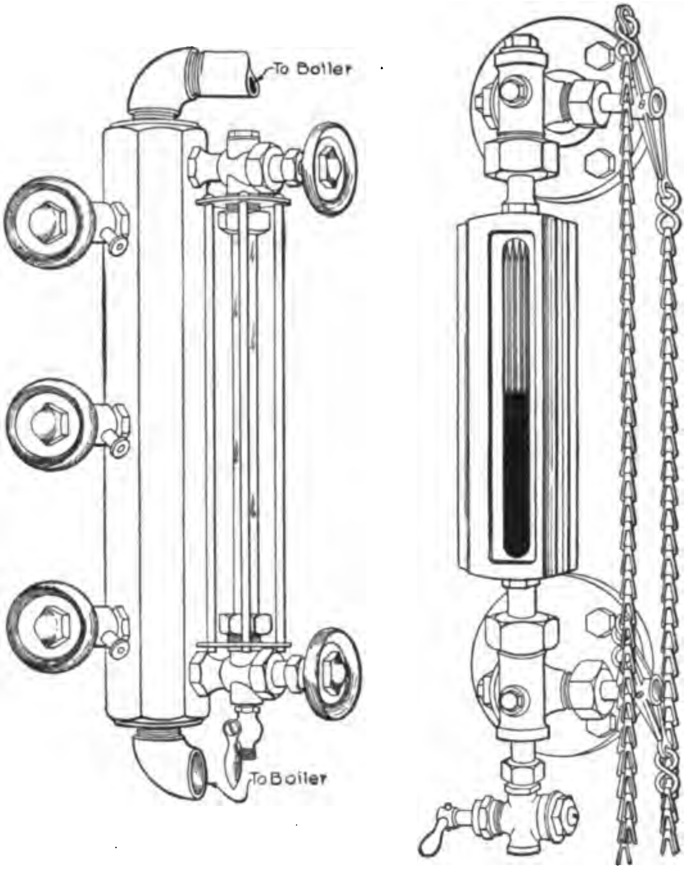


Fig. 35. Ordinary Water Gauge-Glass Supplemented (at right) by "Klinger Patent" Gauge-Glass.

The water gauge is not absolutely reliable, for the water in the gauge, being cooler than that in the boiler, may not indicate the true level, and the small passages leading to it may become choked with sediment. If the gauge-glass is frequently blown out by the engineer and kept clean, this difficulty will be reduced to a minimum.

VALVES

Of all boiler accessories, perhaps the most important are the *cocks* and *valves* by means of which the flow of steam or water may be shut off completely or partially. The valve operates by moving a disc across the pipe in a transverse direction, or by bringing a cap tight upon the seat in a fore-and-aft direction. A cock consists of a block inserted in the passageway, with an opening cut through in one direction. When the handle of the cock is in line of the pipe, the opening allows the steam to pass through; but if turned crosswise, the opening is closed.

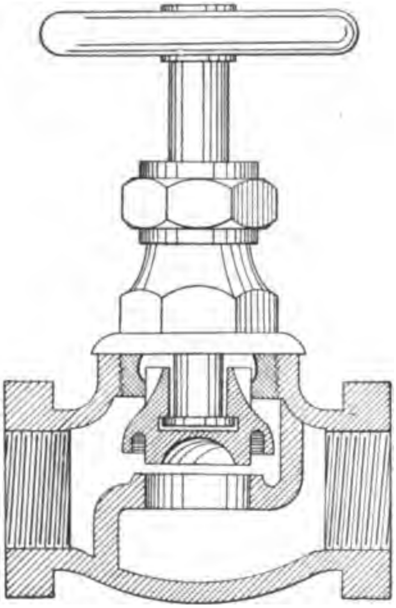


Fig. 36. Ordinary "Competition" Type of Globe Valve.

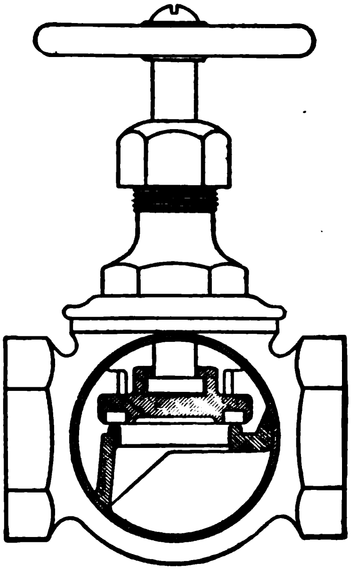


Fig. 37. Globe Valve with Detachable Cap and Removable Interior Disc of Comparatively Soft Material to Insure Tightness.

The Globe Valve. The valve shown in Fig. 36 gets its name from the globular shape of the casing which encloses the valve. Extending across this whole casing is a substantial diaphragm, the central portion of which is in a plane parallel with the length of the pipe. The opening is cut in this portion, horizontal in the figure, through which steam or other fluid may pass when the valve is opened. When the valve is closed, a cap is forced down to close its opening. The rim around the opening is known as the *valve-seat*. The valve-cap is operated by a spindle, which passes through the bonnet of the valve and is mounted at the upper end by a small wheel or handle. To prevent the escape of steam around this spindle, a stuffing-box is provided. The valve-cap may or may not rotate as the spindle turns; usually it does not. The valve shown in Fig. 36 is the ordinary globe valve known to the trade as the "Competition" valve. It is the cheapest valve of the type, and is not satisfactory where absolutely tight work is required. If the cap becomes scored, the valve will leak and is then worthless.

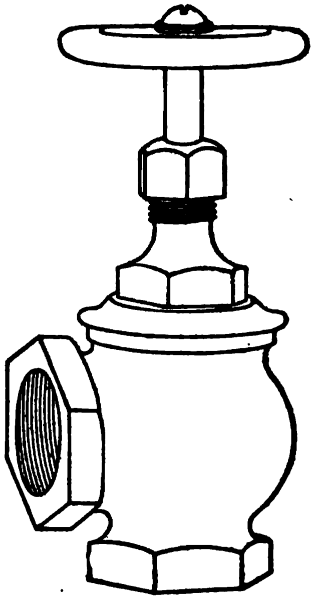


Fig. 38. Angle Valve of Ordinary Globe Pattern.

A valve shown in Fig. 37 has a detachable valve-cap; and instead of relying for tightness upon the valve and seat coming together, metal to metal, a removable disc is provided, which being softer than the metal valve-seat, easily takes up the wear, and the valve not only can be closed tighter, but if anything happens so that the tightness of the valve is impaired, the valve-cap can be replaced by another at a trifling expense. In the cheaper valve, when the cap is scored, the valve is worthless. The valve-seat sometimes has a slight bevel, the valve-cap being shaped like the frustum of a cone.

It is impossible to close a valve tightly if the slightest particle of scale or grit gets between the disc and the seat. If this happens, the valve-seat is likely to become scored so that it does not hold tight; but it may be reground, and if the valve disc itself is damaged, it can readily be replaced.

Angle Valves. An angle valve, shown in Fig. 38, is constructed in a similar manner to the ordinary globe valve, and is sometimes used in place of the straightway valve and an elbow. Both these styles of valve should be so placed in the steam pipe that the entering steam comes beneath the valve-seat. If this is done, the valve-stem may easily be repacked simply by closing the valve. If the steam enters in the opposite direction, a leaky valve-stem cannot be packed, as loosening the stuffing-box would permit the escape of the steam.

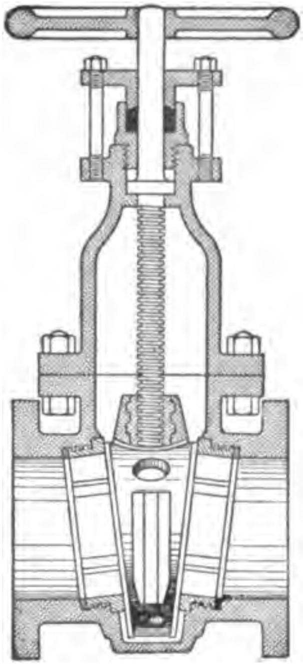


Fig. 39. "Chapman" Gate Valve with Wedge-Shaped Sides.

The Gate Valve. The *gate* or *straightway* valve gives a straight passage through the pipe, and, when open, offers very little resistance to flow. The globe valve, of course, offers much resistance, because the fluid has to change its direction of flow completely.

There are two forms of gate valve—one with wedge-shaped sides, and the other having the valve sides parallel. Fig. 39 shows a "Chapman" valve with wedge-shaped sides. A collar holds the valve spindle at a fixed point; and to open or close, the valve is drawn up or lowered by turning the spindle. When the gate reaches the bottom of the pipe, a wedge on the lower end of the spindle causes the sides to move laterally, with sufficient force to bring a strong pressure against the valve-seat. For heavy work, these valves are made with a rising spindle instead of a stationary one. This possesses the distinct advantage of indicating at a glance whether they are opened or closed; while one cannot tell by looking at the ordinary gate valve whether it is open or not.

Check-Valves. When it is necessary that the flow should always take place in the same direction, as in the feed-pipe of a boiler, *check-valves* are used. There are several forms shown in Fig. 40, one of which has a similar pattern to a globe valve, with a ball or flat valve, the seat being parallel to the direction of flow. The valve is held in place by its own weight, and by the pressure of the fluid in case of a reverse flow. In the *swinging* check-valve, the seat is at an angle of about 45 degrees to the direction of flow. It is fitted somewhat loosely where it is fastened to the swinging arm, so that it may properly seat itself. This form is usually preferred, as it offers less resistance to flow and there is less chance for impurities to lodge on the valve-seat. When a check-valve is used in the boiler-feed pipe, there should be a *stop-valve* between it and the boiler, which can be shut in case the check-valve should get out of order.

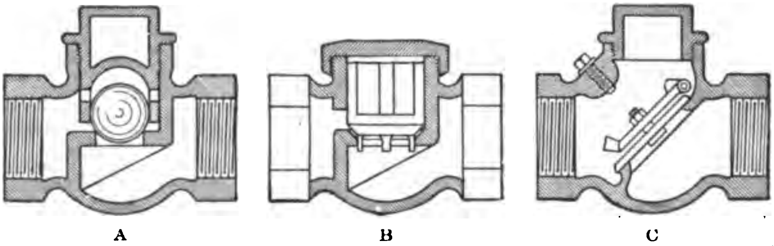


Fig. 40. Types of Check-Valves. *A*-Ball-Valve; *B*-Flat Valve; *C*-Swinging Check-Valve.

Materials. For pressures under 200 lbs. per square inch, cast iron may be used for the body of the valve; but, for economy, it should be used only when the pressure is over 130 lbs. For heavy work it is frequently necessary to have a massive valve that cannot easily be broken. In such a case a cast-iron body is the most suitable thing. The valve-seat, valves, spindles, stuffing-box, glands, and nuts are usually made of gun-metal or brass. For very high pressures, especially on steam mains, cast steel is generally used, with gun-metal fittings similar to those enumerated for the cast-iron valves.

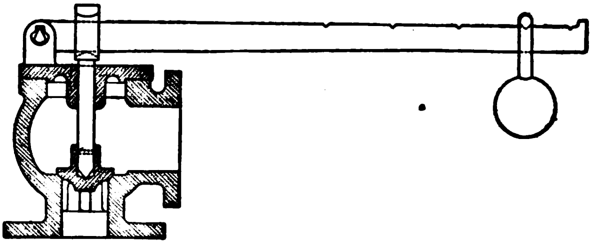


Fig. 41. Common Type of Lever Safety-Valve.

Safety-Valves. Safety-valves are used for reducing the pressure in the boiler when it exceeds a certain limit, and to give warning of high pressure. There are several different types, but the essential features are a valve opening upward, held on its seat by a weight or spring. When the pressure in the boiler exerts a force greater than that holding down the valve, the valve will open automatically.

The *lever safety-valve* shown in Fig. 41 is the most common type for stationary work, especially for small boilers. The valve is held in place by a weight at the end of a lever. The force required to lift the valve is governed by the location of the weight on the lever-arm. The body of the valve is usually made of cast iron, the seat being of brass. An opening on the side of the valve may be connected with the feed-water heater or drain, if the escape of steam into the air is undesirable. If the valve becomes leaky, it should be reground; but no attempt should be made to make it tight by increasing or moving the weight on the lever.

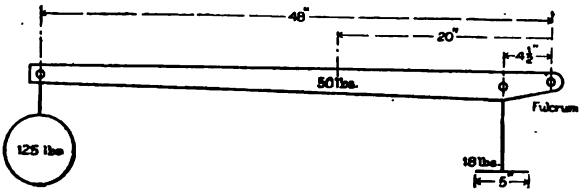


Fig. 42. Diagram for Safety-Valve Calculations.

The amount of necessary weight on the lever, and its distance from the fulcrum, can be determined in the usual manner of computing leverage forces and moments, remembering that weight times weight-arm is equal to power times power-arm. In such a valve as this, *power* is the steam pressure, and the *power-arm* is the distance of the center of the valve from the fulcrum. There are four weights acting downward—the ball, the lever-arm, the valve, and the spindle—and in the process of computation the weight and leverage of each must be taken into account.

Suppose, for example, that we have a lever safety-valve such as is illustrated in outline in Fig. 42, and that we know the following conditions: the ball weighs 125 lbs., and is suspended at the end of the lever 48 inches from the fulcrum; the valve and valve spindle together weigh 18 lbs., and are 4½ inches from the fulcrum; the lever-arm itself weighs 50 lbs. If the valve-seat is 5 inches in diameter, at what pressure will the valve blow off, ignoring the friction of the stuffing-box and fulcrum pivot?

The center of gravity of the lever-arm must be determined from the drawing (Fig. 42), and this is found to be 20 inches from the fulcrum. The leverage of the weights acting downwards is then as follows:

Ball	125 × 48	= 6,000
Lever	50 × 20	= 1,000
Valve and Stem	18 × 4½	= 81
Total moment		= 7,081 inch-pounds.

Now, if the valve-seat diameter is 5 inches, the area of the valve will be $\frac{\pi D^2}{4} = \frac{3.1416 \times 25}{4} = 19.63$ sq. in. The total moment to be overcome is 7,081 inch-pounds, and its distance from the fulcrum is 4½ inches. Therefore the necessary upward pressure on the valve will be $\frac{7{,}081}{4\frac{1}{2}} = 1{,}573.5$ lbs. If the area of the valve is 19.63 sq. in., then the necessary pressure in pounds per square inch would be $\frac{1{,}573.5}{19.63} = 80$ lbs., approximately. That is, this safety-valve would blow off when the boiler pressure reached 80 lbs. per square inch.

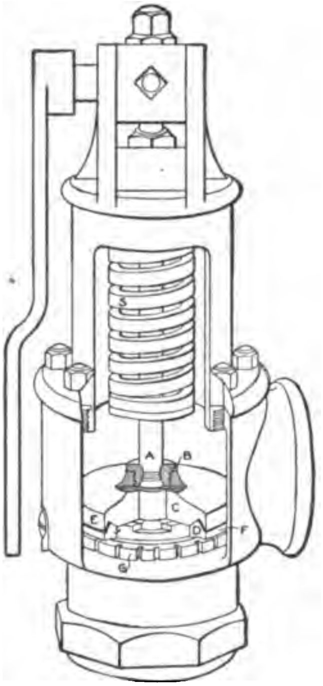


Fig. 43. "Crosby" Pop Safety-Valve for Stationary Boilers.

If it is desired to design a valve that will blow off at known pressure, the same principles will apply, but the computations will be figured in the reverse order. The area of the valve, times the boiler pressure, would give the total lifting force; and this, multiplied by its leverage, would give the lifting moment, which would be resisted by the downward moment of the combined weights of valve, valve-stem, lever, and ball. If the moments of the lever, valve, and valve-stem were known, the rest, of course, would be made up by the ball. If the length of the lever-arm were known, then the weight of the ball would be varied to correspond; and, conversely, if the weight of the ball were fixed, the length of the lever must be made to correspond.

The lever safety-valve has several defects. It does not close promptly when the pressure is reduced; and it is likely to leak after it is closed, and may readily be overloaded, or even wedged on its seat. It is essential that a safety-valve should be automatic, certain in its action, and prompt in opening and closing at the required pressure. It must be one that can be relied upon under all circumstances.

The *pop safety-valve* fulfils the above requirements better than those of the lever type. Pop valves open when the steam pressure is sufficient to overcome the tension of the spring. Fig. 43 shows a "Crosby" pop safety-valve for stationary service. The valve *C* is connected by the flange *B* to the central spindle *A*, and is held down on its seat by the pressure of the spring *S*. The valve *C* is provided with wing guides and an annular lip *E*. The guides fit smoothly into the seating *D*, upon which the valve rests. The seats of the valve have an angle of 45 degrees. The under face of the lip *E*, together with the seating, forms a small chamber through which all the steam must pass to the open air. A number of small holes drilled vertically through the flange *F*, connect with the chamber and allow part of the steam to escape. The action of the valve is regulated by the screw ring *G*, which allows more or less steam to escape through the holes in the flange *F*. Raising the screw diminishes, and lowering it increases, the area of the holes. If the loss of steam is too great when the valve blows, turn the screw ring down.

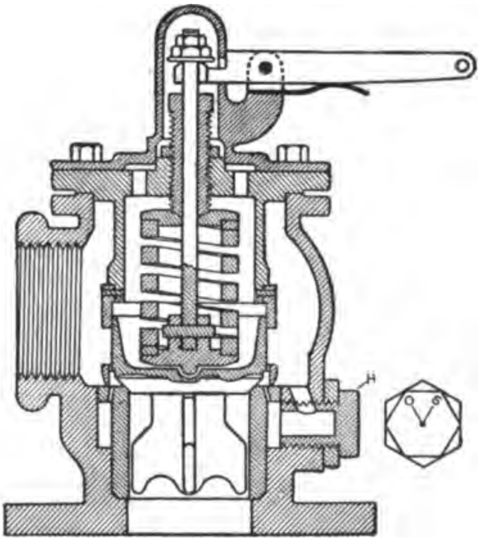


Fig. 44. "Ashton" Valve with Pop Regulator for Stationary Boilers.

Safety-valves *should be connected directly to the boiler* without any pipe or elbow. They should be tried every day by means of the lever.

The valve shown in Fig. 44 for stationary boilers, is made by the Ashton Valve Company. The general principles are those of all pop safety-valves. The valve-seat is made of composition or nickel, and with a bevel of 45 degrees, as is the United States Government standard. The pop chamber is surrounded by a knife-edge lip, which wears down in proportion with the seat, thus keeping the outlet of the same relative proportions, giving a constant amount of pop.

The amount of pop—that is, the difference of pressure between the opening and the closing of the valve—is regulated from the outside by means of the screw-plug *pop regulator* shown at H in Fig. 44. If more pop is desired, turn the regulator so that S will be more nearly perpendicular. To lessen pop, make O more nearly perpendicular. The springs are made of Jessop's best steel.

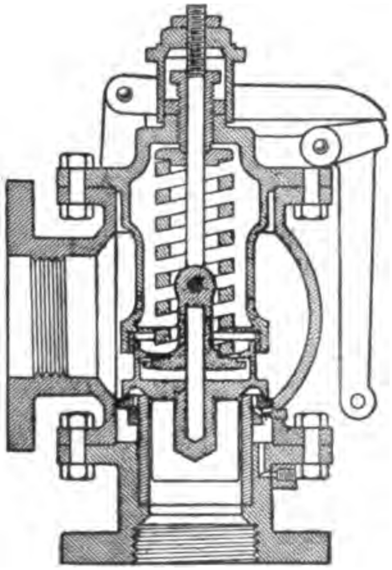


Fig. 45. "Star Marine" Pop Safety-Valve, with Cam Lever.

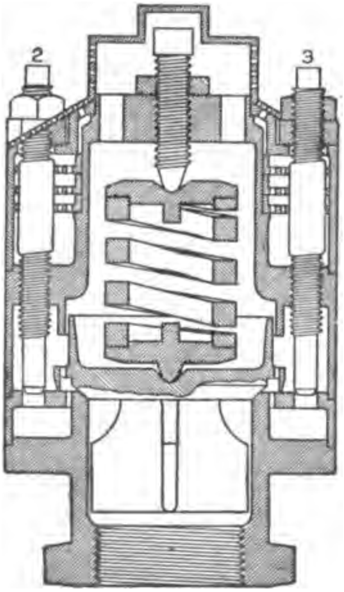


Fig. 46. "Ashton" Safety-Valve for Locomotive Boilers, with Pop Regulators on Each Side, and Top Muffler.

The inlet and outlet are both on the same casting, so that the valve may be taken apart to be cleaned or repaired, without disturbing the boiler connection. It has a lock-up attachment, so that the regulating parts cannot be tampered with, either by accident or by design. The spring is encased, thus protecting it from the steam.

The "Star Marine" pop safety-valve is shown in Fig. 45. It has a bevel seat, and is provided with a cam lever by which it may be raised from its seat when there is no steam pressure. The outlet of the valve, if desired, may be piped to the supply tank or to any other point.

Safety-valves for *locomotive boilers* must be made of heavy material to stand the severe usage. They should be so constructed that they will not cock or tilt. The "Ashton" valve shown in Fig. 46 is constructed so that the amount of pop can be regulated by merely turning the two posts marked 2 and 3 to the right or left. The noise of the steam escaping from the ordinary safety-valve is disagreeable, and in some States the law requires the use of the *muffler safety-valve.* The Ashton valve shown in Fig. 46 has a top muffler.

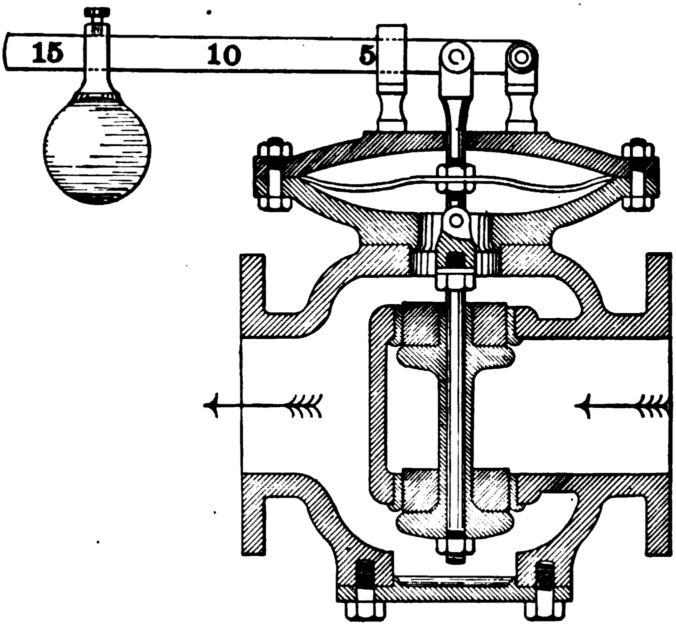


Fig. 47. "Holt" Reducing Valve with Diaphragm Regulating Pressure.

Reducing Valves. Sometimes steam is desired at a lower pressure than that of the boiler. For instance, a small low-pressure engine may be run by steam taken from the same boiler that supplies a higher-pressure engine. This reduction is accomplished by throttling the steam by means of *reducing valves.* These are arranged to be operated automatically so that the pressure can be reduced and a constant pressure in the steam pipes maintained. There are several forms in general use.

In the "Holt" valve, Fig. 47, the low-pressure steam acts on the lower side of the diaphragm; and the weight, which may be set so as to cause the desired pressure, acts on the other. The movement of this diaphragm causes a balanced valve to move to or from its seat. The valve opens until the steam pressure equals the weight above. The pressure in the main steam pipe does not affect the movement of the valve. It depends only upon the pressure on the two sides of the diaphragm.

Another form, the "Mason," is shown in Fig. 48. A spring, which may have its tension altered by a key, takes the place of the lever and weight in the Holt valve. When the pressure in the low-pressure system has risen to the required point, which is determined by the spring, the valve closes, and no more steam is admitted until the pressure falls sufficiently to open the valve again.

In another form, a piston acted on by the low-pressure steam regulates the opening of a balanced valve, and this maintains a constant steam pressure.

In the "Foster" reducing valve, the valve is held open by the spring and levers, until the steam pressure at exit presses on the diaphragm sufficiently to close the valve. The valve is held open so as to admit just the proper amount of steam to maintain the required pressure.

When a reducing valve is used, a stop-valve should be put in to prevent flow when steam is not in use.

BLOW-OUT APPARATUS

Boiler feed-water, if taken from rivers or ponds, is likely to contain vegetable matter as well as solid materials. The vegetable matter will usually float to the surface, while the solids will collect at the bottom. To keep the boiler clear of such sediment; it is necessary to provide two *blow-outs*—a *surface* blow-out, to take care of what rises to the top; and a *bottom* blow-out, to take out the sediment that collects at the bottom of the boiler. The surface blow-out usually consists of a dish or funnel-shaped receptacle set with its face vertical, as shown in Fig. 49. When the water-level is in line with this blow-out opening, the opening of the valve at the bottom will skim the impurities from the surface of the water quite readily. Oil may get into the boiler through the feed-water, and a considerable portion of it can be removed in this manner.

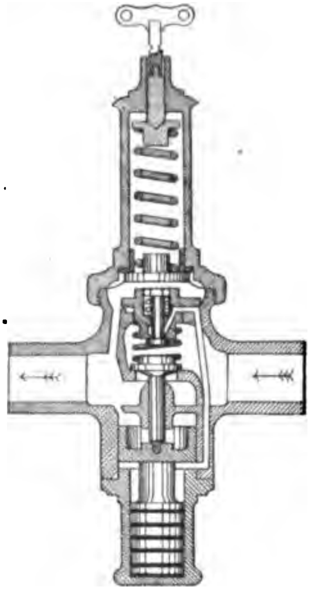


Fig. 48. "Mason" Reducing Valve. Pressure Regulated by Means of a Spring.

The bottom blow-out consists merely of a pipe leading from the bottom of the boiler outward. Both these blow-outs may be connected into one outlet.

In water-tube boilers a *mud-drum* is usually installed, which readily collects the solid matter, and the bottom blow-out is then connected with this mud-drum. Fig. 50 shows an arrangement of surface and bottom blow-outs as usually installed on a Scotch boiler of the marine type.

If the feed-water contains salt, which may frequently happen in marine practice, it is necessary that the boiler should frequently be blown out in order to remove the excess of salt. The density of the boiler water, if salt feed is used, should be carefully determined by a salimeter. The loss due to this frequent blowing out is considerable, as a large amount of heat is necessarily wasted; but it cannot be avoided, except by the use of fresh water, which sometimes may be impossible at sea.

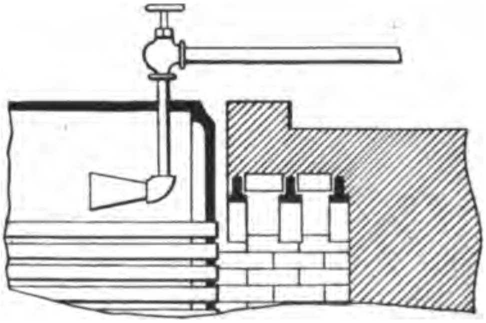


Fig. 49. Surface Blow-Out Installed in Boiler.

The blow-out pipe leading from the bottom of an externally-fired boiler through the brick setting, if not properly protected, may be burned off, owing to the heat of the fire. This pipe is frequently covered with asbestos or other fire-resisting material;

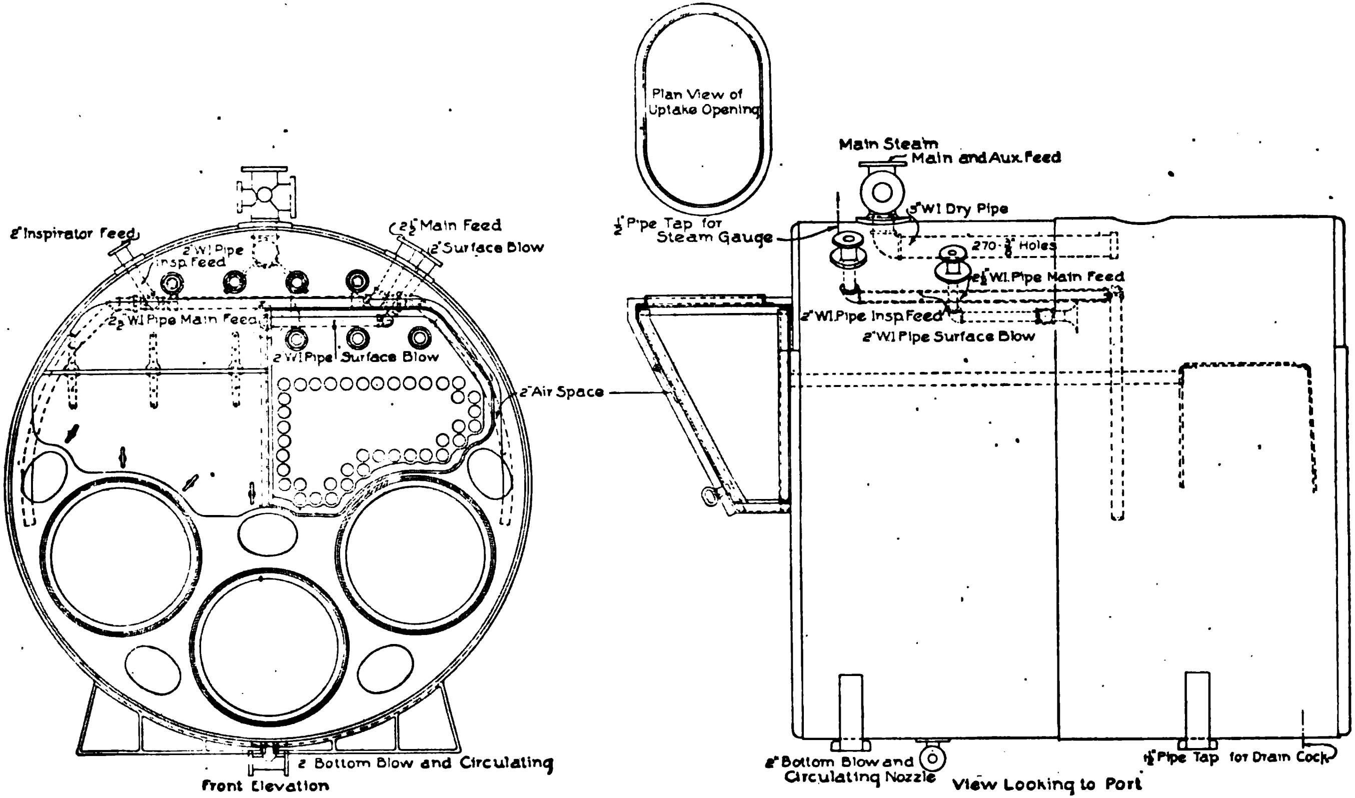


Fig 50. Scotch Boiler of Marine Type, Showing Arrangement of Surface and Bottom Blow-Outs.

but it can be best protected by the means shown in Fig. 51. A pipe connected to the boiler slightly below the water-level, runs out through the brick setting and connects into the main blow-out pipe. This causes a circulation of water continually to pass through the system, and prevents destruction of the blow-out pipe. When it is necessary to use the bottom blow-out, the valve *A* is closed, and the blow-off valve *B* is opened; otherwise, *B* is closed, and *A* is open while the water circulates..

The blow-out pipe is usually shut off by a cock, which, although not so easily operated as a valve, is more trustworthy. Frequently both a cock and a valve are provided. Should a small particle of sediment lodge on the valve-seat, it would be impossible to close the valve tightly, and a considerable leakage would result, while an inspection of the valve would not indicate whether it were completely closed or not. But a glance reveals the fact whether or not a cock is tightly closed. The cock is likely to stick because of corrosion or unequal expansion, but, if frequently opened, this difficulty is not of great weight. The plug and casing of the cock should not be made of the same material, as in that case they will more readily stick if the cock remains closed any length of time.

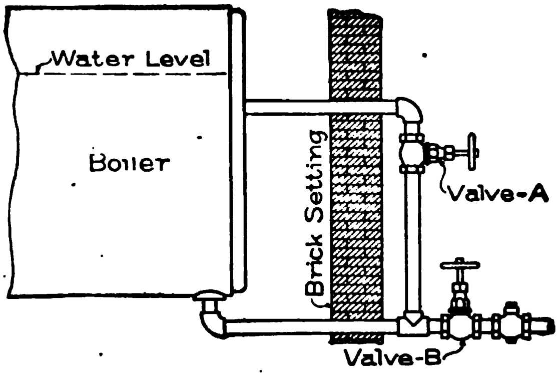


Fig. 51. Method of Protecting Bottom Blow-Out Pipe by Means of Circulation Pipe Connected to Boiler.

FEED APPARATUS

Perhaps the most important of all auxiliaries connected with the boiler is the feed apparatus. This is vital; for, if the feed is interrupted and the water runs low in the boiler, not only is there danger of damaging the boiler itself, but a disaster may follow of far greater concern. For marine purposes—and the same is true to a considerable extent in stationary work—at least two independent feed systems should be provided. In marine work, the main feed-pump draws water from the filter box or feed-water heater, and pumps it into the boilers under ordinary conditions. There should be a by-pass around this pump, and the feed line should be connected by means of a valve to what is known as the *donkey pump*, which may be used for auxiliary feed purposes in case the main pump is damaged or needs repairs in any way.

Both these pumps draw from and discharge into the same feed line; but, to provide against emergencies, there is usually a cross-connection to the sea, so that sea water may be had if necessary. While in port, when the main engines are not running, and consequently when the feed-water cannot be heated economically, an injector is almost invariably used. On land it is usually considered sufficient to install an injector in addition to the feed pump, although in large plants an auxiliary feed pump should be installed as well. In a small plant the fireman usually attends to the water; but on board ship and in large plants, a water tender is usually provided, whose business it is to keep the water in the boiler at the proper level. His task may be materially lessened by some automatic arrangement, so that if the water discharged into the hot well from the condenser rises above the normal level, a float will open the valve leading to the feed-pump and increase the rapidity of its stroke. This will reduce the level of the hot well or filter box, as the case may be.

Such an arrangement as this will keep a fairly uniform level of water in the boilers; and if a surface condenser is employed, and all the condensation is pumped back into the boilers, the water-level will remain constant except for slight leakages of steam and for the possibility of improper action of the feed-pump. Leakage of steam can be made up from the supply of fresh water. At sea, salt water may have to be used for this purpose although its use is objectionable.

There is a considerable difference of opinion as to where the feed-water should be introduced into the boiler, although the consensus of opinion seems to be that it should enter not far from the water-line. In stationary practice, the feed-water is introduced at the rear of the boiler near the bottom; but this is open to grave objections, for the feed-water, being comparatively cool and being introduced into the coldest part of the boiler, naturally tends to become dead water and to retard proper circulation which is essential to economical steaming and often essential to the safety of the boiler itself.

The best place for introducing the feed-water will naturally depend upon the type of boiler, and the service for which it is intended. If the entering water is of high temperature, it might enter near the bottom of the boiler. But if the feed-water is comparatively cold — and it is always colder than the water in the boiler and the surrounding steam if the circulation is good—great care must be taken that it does not strike directly against the hot boiler-plates, as it might thereby cause local contraction and possibly a serious leak, and it should be introduced in such a way as to make sure of its aiding the natural circulation of the boiler.

The higher the steam pressure in the boiler, the more difficult becomes the problem of feed, and the more danger there is of injury to the boiler by the comparatively cold feed-water striking hot plates. It is a universal practice in marine work, and a common practice on land, especially for internally-fired boilers, to cause the feed to enter above the water-level near the center of the boiler; then branching off into two pipes, one leading to each side through the steam space until the side of the boiler is reached; and then running downward toward the bottom. The feed-water, which very likely has been previously heated by a feed-water heater, is still further heated by its passage through this feed-pipe, which is in direct contact with the live steam of the boiler. This internal feed-pipe, turning down at the sides, causes the water to strike the outer shell of the boiler which is the most remote from the fire, and this downward motion materially assists the circulation in the boiler. When this arrangement of feed is adopted (see Fig. 50), care must be taken that the lower end of the feed-pipe is well below the low-water level. If the end of the pipe is alternately immersed in water and then exposed to steam, violent explosions in the pipe are likely to follow, although they are likely to do nothing more serious than break an elbow or frighten the attendants.

In stationary practice, it is quite common to admit the feed-water into the steam space through a horizontal pipe entering it through the tube-plate a few inches below the low-water level, and terminating in a perforated pipe of large diameter. This method distributes the feed-water admirably, and allows it to become considerably heated before it reaches the bottom of the boiler. If the feed-water contains a considerable amount of magnesia or calcium carbonate, holes so arranged in the feed-pipe are likely to become clogged and the feed interrupted. Water of this sort should be fed into a trough, or the feed-pipe be opened at the top by a long slot, so that the feed-water may overflow. The trough in this case forms an admirable mud-drum or sediment collector.

In internally-fired boilers of the "Cornish" or "Lancashire" type, the feed is usually delivered near the bottom through a horizontal pipe—either through the front end or by a vertical pipe through the crown. This method is not conducive to the best circulation.

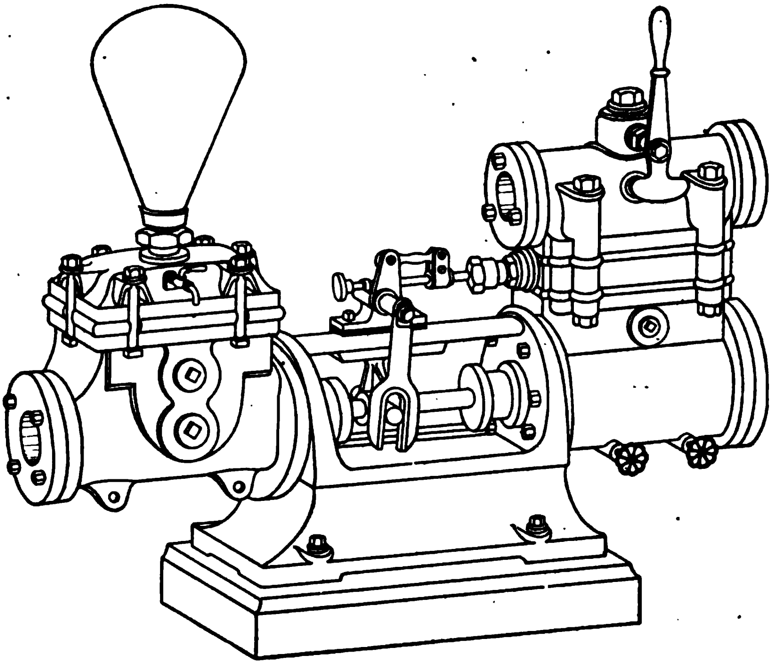


Fig. 52. Steam-Driven Boiler Feed-Pump.

In addition to these effects on circulation, there are other grave objections to introducing feed-water near the bottom of the boiler; for, should anything happen to the feed-pump, or a piece of scale lodge under the check-valve, the water might be almost entirely blown out of the boiler before the difficulty could be discovered or remedied.

If the pipe enters in the vicinity of the low-water level, no water could be drawn out below this point.

The feed supply should always be regulated so as to keep the water-level as nearly stationary as possible; this is not only much more economical, but also far better for the boiler, than to wait for the water-level to fall and then feed a few inches rapidly. The sudden introduction of a large volume of comparatively cold feed-water, causes local contraction of the plates, and hence tends to cause leakage; moreover, it necessitates irregular firing if anything like a uniform steam pressure is to be maintained.

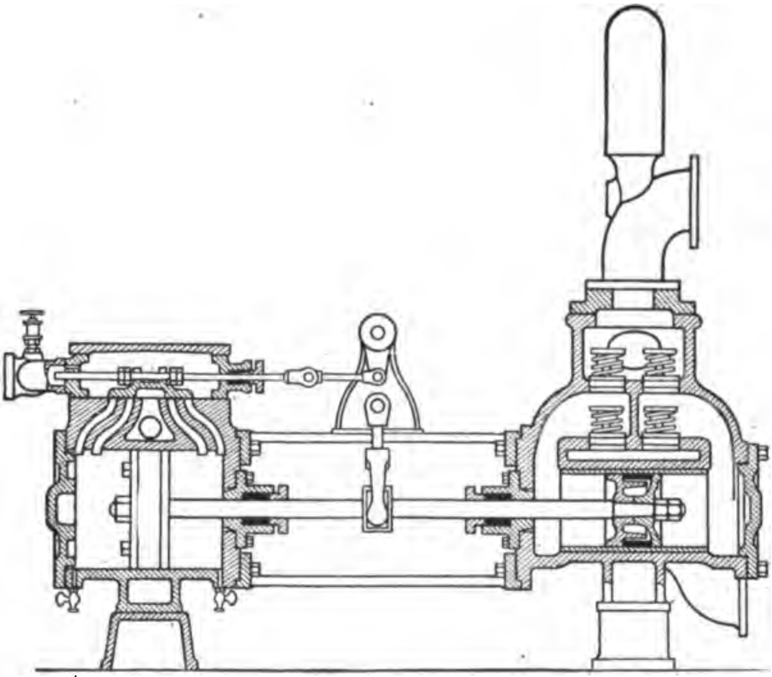


Fig. 53. Section of "Worthington" Duplex Steam Pump

Sometimes the feed-water is forced into the steam space in the form of a fine spray. In this way it not only is thoroughly heated before mingling with the water in the boiler, but the air is got rid of; and salts, such as sulphate of lime, insoluble at high temperatures, are immediately precipitated. But the advantage of introducing the feed-water in a body so as to produce useful circulating currents, should not be overlooked.

If several boilers are attached together in the form of a battery, each should be supplied with an independent connection to feed-pipe. Otherwise a damage to the feed-pipe in one boiler might affect the others. Moreover, if several boilers are fed from one pipe, the pressure in each of them being slightly different, an excess of water will naturally be fed into the boiler having the *least* pressure, whereas it is usually the case that the most water is needed in the boiler having the *greatest* pressure. The automatic float previously referred to, can regulate the amount of water fed into boilers only in a general way, through providing a method by means of which all the condensation is fed back into some of the boilers; but the quantity of feed which is led into each individual boiler must be watched and regulated by the water tender, who can open or close the individual valves as desired.

Pumps. Boilers are usually fed by a small, direct-acting *steam pump* placed near the boiler. Although these pumps require a large steam supply per horse-power per hour, the total amount of steam used is small because the work done is small A more economical pump is the power pump driven by the large steam engine; but in this case the rate at which water is supplied is not easily regulated to the demand of the boiler. Power pumps are usually arranged to pump a larger quantity of water into the boiler than is required, the excess of water being allowed to flow back into the suction pipe through a relief valve.

The pump shown in Fig. 52 is well adapted for feeding boilers. In Fig. 53 is shown the section of a duplex "Worthington" steam pump. The action of each of these two types is similar. Steam, controlled by valves, drives the piston in the steam cylinder, which moves the plunger in the water cylinder, since both are fastened to the same rod. The movement of the plunger forces a part of the water in front of it up through the valves into the air-chamber, and through the pipes into the boiler. On account of the partial vacuum caused by the movement of the plunger, water will be drawn from the suction pipe, through the valves, into the pump cylinder, filling the space left by the movement of the plunger. During the return stroke, this water is forced up into the air-chamber, and a like quantity enters the other end of the pump cylinder. The valves are kept on the seats by light springs, until the pressure on the bottom side is sufficient to lift them and allow water to flow through.

When two pumps are placed side by side, and have a common delivery pipe, the machine is called a *duplex pump*. It is usual to set the steam valves so that when one piston is at the end, the other is at the middle of its stroke. A duplex pump having a large air-chamber and valves set to act in this manner, delivers water with an approximately constant velocity.

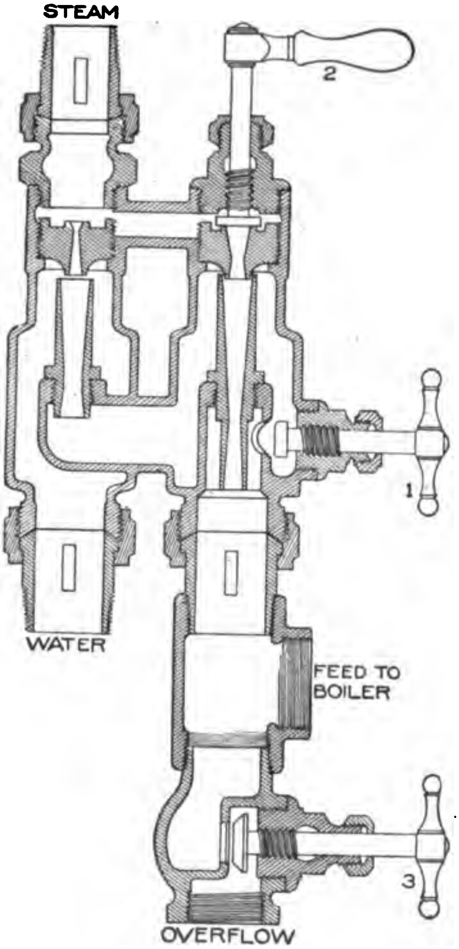


Fig. 54. Sectional View of "Hancock" Injector.

Injectors. Water may be forced into a boiler by an *injector* or *inspirator*. By means of this instrument, the energy of a jet of steam is used to force the water into the boiler. That there is sufficient energy to do this work is evident from the fact that each pound of steam, in condensing, gives up about 1,000 B. T. U., and a B. T. U. is equivalent to 778 foot-pounds. Not all the energy of the jet of steam is used in forcing water into the boiler; some is wasted, and much is used to heat the feed-water.

The action of the injector is briefly as follows: The steam escapes from the boiler with great velocity, and, as it passes through the cone-shaped passage, draws air along with it, thus creating a partial vacuum in the suction pipe. Atmospheric pressure forces water up into the suction pipe, and the jet of steam which it meets is partly condensed.

The energy of the jet carries the water along with it into the boiler.

Experiments show that the injector, if considered as a pump, has a very low efficiency. When used for feeding a boiler, it has a thermal efficiency of nearly 100 per cent, since all the heat of the steam passes to the water except the slight amount lost in radiation. The pump, however, has one great advantage over the injector; it can force hot water from a heater into the boiler, while an injector can be used only with cold or moderately warm water.

Figs. 54 and 55 show the interior section and exterior of a "Hancock" inspirator. To inject water to the boiler, first open overflow valves 1 and 3; close valve 2; and open starting valve in the steam pipe. When the water appears at the overflow, open 2 one quarter-turn, close 1, and then close 3. The inspirator will then be in operation. When the inspirator is not working, open both 1 and 3 to allow water to drain from it.

Both temperature and quantity of delivery water can be varied by increasing or decreasing the water supply. When the water in the suction pipe is hot, either cool off both pipe and injector with cold water, or pump out the hot water by opening and closing the starting valve suddenly.

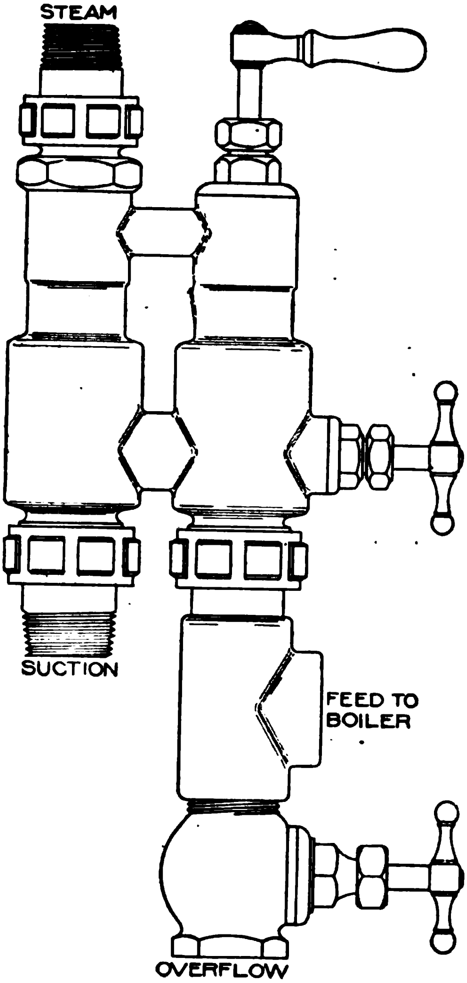


Fig. 55. Exterior View of "Hancock" Injector.

Circulating Apparatus. There is always more or less danger in starting a fire under a boiler. If the circulation is poor, the result will be that not only will the water be of an uneven temperature, hot near the top and cold at the bottom, but the boiler shell is likely to be subjected to severe strain, owing to the difference of temperature arising from the stagnation of the cold water near the bottom. The fire must be started slowly, and a considerable time consumed in getting up steam. To overcome the difficulty of poor circulation, several mechanical devices have been applied.

The first device tried was a *hydro-kineter*—a sort of injector—in which jets of steam driven through a conical nozzle drew in the surrounding water. This was so arranged as to induce the cold water to flow from the bottom toward the top, where it was more intensely heated. This arrangement is efficient, but slow of action. In large marine boilers—in which the fire is cautiously started, as is proper—the temperature at the surface of the water, four hours after lighting up, has been found to be as high as 205°, while at the bottom it was only 73°. Several observations with a hydro-kineter in action have shown the temperatures to be 205° and 144° respectively. It was six hours more before the temperature was equalized throughout.

In naval vessels, where it is frequently necessary to raise steam rapidly, this device is altogether too slow. It has, moreover, two other drawbacks. There must be an auxiliary boiler under steam pressure, and it will cease to act when the temperature and pressure of steam in the main boiler has reached that in the auxiliary boiler. The steam jet, in the American Navy, has been replaced by a jet of feed-water forced through a conical nozzle. This arrangement answers very well so long as steam is being drawn from the boiler; but when the boiler is at rest and steam is being raised, it is inoperative.

The best service can be had by means of small centrifugal pumps fixed beside the boilers, which take water from the bottom of the boilers and discharge it a little below the water-level. The pumps may be turned by hand while raising pressure, and may be worked by steam when sufficient pressure has been attained. A small engine of perhaps 1½ horse-power is sufficient to give a proper circulation to a large boiler. With such a circulating device, steam can be raised with safety, in a comparatively short time.

Evaporators. No engine can be run without a certain loss of water, due either to a slight continuous leakage or to blowing off. In stationary practice, this loss can be readily made up by the application of fresh water; but at sea it is seldom possible to carry a sufficient amount of fresh water, and the make-up must be had either from sea water, or from fresh water provided by the use of an *evaporator*. The evaporator is really a small boiler, the water in which is heated by a steam coil supplied from the main boiler. The evaporated water—called the *evaporation*—passes into the condenser and then becomes a part of the regular feed water.

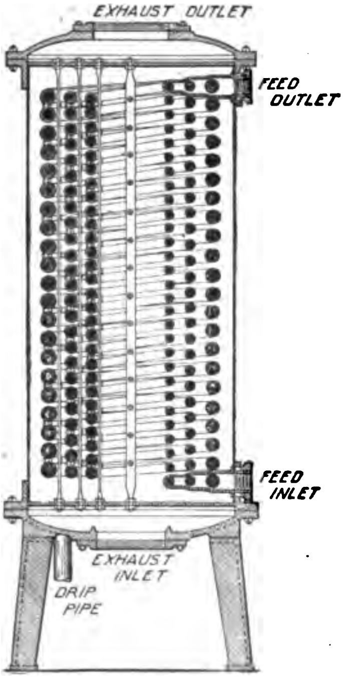


Fig. 56. Feed-Water Heater. Closed Type.

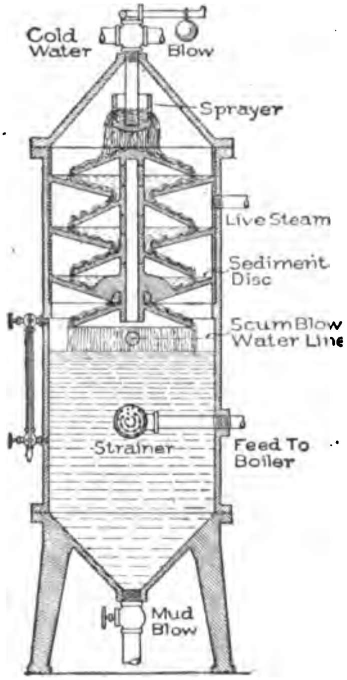


Fig. 57. Feed-Water Heater, Open Type.

In a single evaporator, if the evaporation passes directly to the condenser, its heat is lost to useful work. To provide a more economical arrangement, multiple evaporators are installed, which consist of a series, the evaporation from the first passing into a coil in the bottom of the second; the water in the second condenses the evaporation from the first, while at the same time the evaporation from the first helps to heat the water of the second. The steam and water pass through the series of heaters in opposite directions.

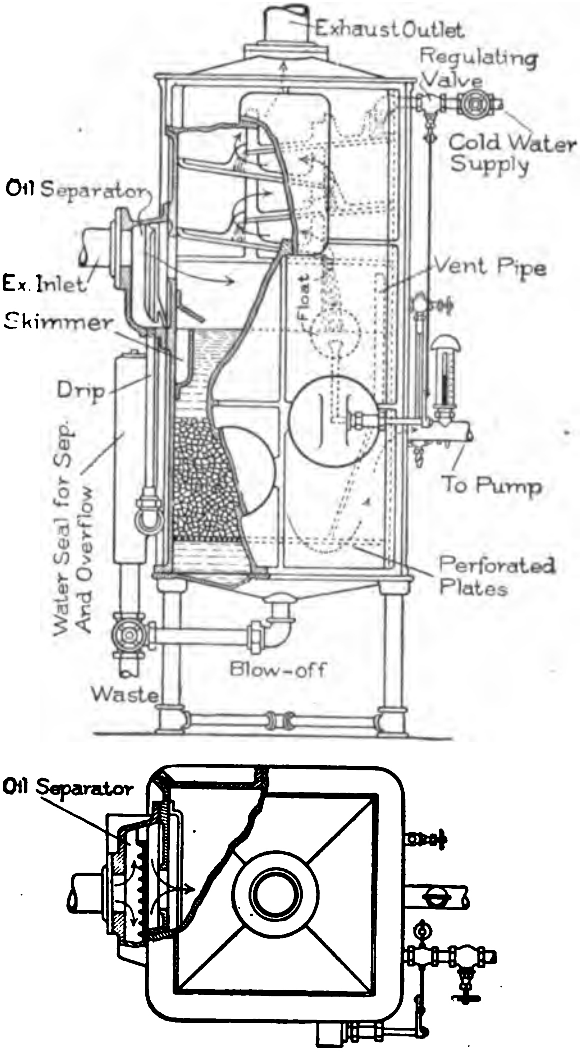


Fig. 58. "Cochrane" Combined Feed-Water Heater and Purifier, Open-Heater Type.

It is a rule in the French Navy, to provide 380 lbs. of fresh water per hour for each 1,000 indicated horse-power; this provides for a loss of about 2 per cent without drawing on the reserve supply, which is 4,500 lbs. for the same amount of power.

The evaporator may be arranged to communicate with a low-pressure valve-chest, in which case the evaporation may be made to do work in a low-pressure cylinder of a triple-expansion engine before entering the condenser, or it may be connected with the feed-water heater if the exhaust steam is inadequate.

Feed-Water Heaters. The introduction of feed-water at a high temperature increases the economy and tends to prolong the life of the boiler. The injurious effects from unequal expansion are diminished; and when the feed is warmed by exhaust steam or by the waste gases in the uptake, the saving of fuel is considerable.

If this gain comes from waste gases or exhaust steam, which would otherwise make no return for their heat, the gain is clear; but there is no gain in thermal economy by heating feed-water with live steam directly from the boiler.

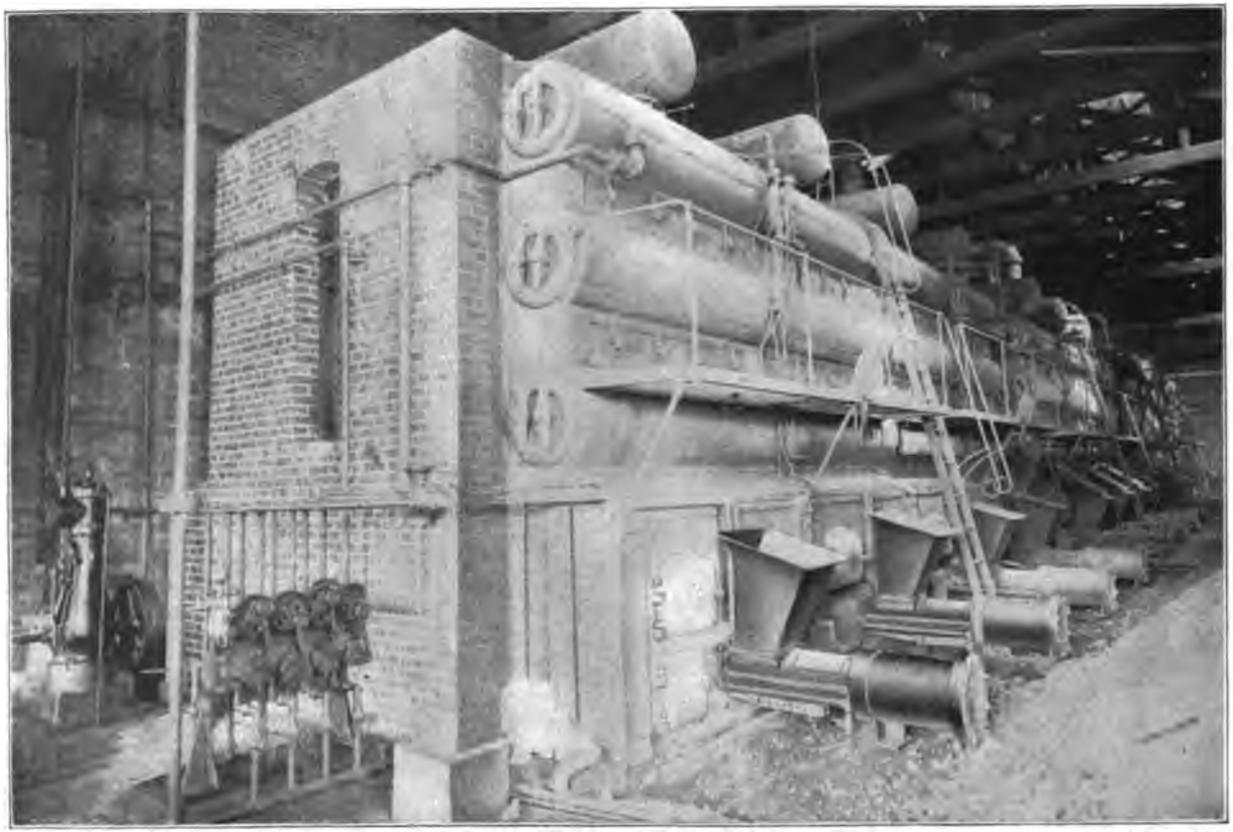
There are several ways of heating the feed-water. In condensing engines, the feed-pump discharges from the condenser into the hot well, and the water is drawn from the hot well at a temperature of 100° to 140° F. This, however, if the pressure is over 100 lbs., is entirely inadequate; and for the best economy, feed-water at this temperature should be passed through some form of feed-water heater. In the non-condensing engines, it is absolutely necessary that in some way the feed-water should be heated by the exhaust steam or by waste gases from the chimney, the apparatus in the first case being called a *feed-water heater*, and in the second an *economizer*.

The feed-water heater may be arranged so that it will not only heat the water, but will at the same time purify it, precipitating the calcium and magnesia salts, which collect on suitably prepared plates, and gathering, at the bottom of the heater, dirt and other sediment that would injure the boiler.

There are two types of feed-water heater—the *open*, which is frequently used in land work; and the *closed*, which may be used either on land or at sea. In the open heater, the steam raises the temperature of the water by mingling with it in direct contact. The closed type of heater resembles in its action a surface condenser; the steam used for heating purposes surrounds tubes which contain the feed-water, or the water circulates about tubes through which the heating steam passes.

Fig. 56 shows a feed-water heater of the closed type, the exhaust steam heating the feed-water within the tubes. The heater shown in Fig. 57 is of the open type, the feed-water becoming heated and depositing sediment while flowing from one tray to another.

The "Cochrane" heater, Fig. 58, is a combined heater and purifier of the open-heater type, the water entering at the top and flowing in a thin sheet over a series of trays. The exhaust steam enters through the oil separator, and rising among the trays, heats the water to about 210° F, the action being similar to that of a jet condenser.



JONES UNDERFEED STOKERS IN STATION OF THE TOKYO ELECTRIC CO.,
TOKYO, JAPAN.

BOILER ACCESSORIES

PART II

STEAM SEPARATORS

Priming. Steam is said to be *wet* or to be *superheated*, according as it has an excess of moisture or an excess of heat. Wet steam not only is uneconomical, because it carries a considerable amount of heat into the engine in the form of water, which cannot do useful work, but, if a considerable amount of water gets into the engine, it is really dangerous, for it may so completely fill the clearances that the piston will strike a blow against the cylinder-head sufficient to break it. The water in the pipes, moreover, may cause a serious hammering, which not only is exceedingly annoying, but may be actually dangerous, for a severe water-hammer may break the joints of the steam pipes, and a considerable quantity of escaping steam at high pressure would be exceedingly dangerous to the lives of the engine-room attendants. This especially would be true on board ship, where the engine-room is small, the supply of air meager, and the means of escape limited.

A considerable amount of water may be deposited in a sag in the pipe line, and would undoubtedly remain there for a considerable length of time if the pressure in the boiler did not fluctuate; but a sudden rise of boiler pressure would likely cause this water to pass bodily through the pipe toward the engine. Moisture is carried directly from the boiler as a result of *priming*. This is caused by steam bubbles which, instead of bursting, become connected on the surface of the water, forming a foam, half-liquid, half-gaseous, which fills the steam space and passes out of the steam pipe. Priming may be due to fluctuations of boiler pressure or to the presence of dirt, oil, or other foreign matter. The smaller the free surface of the water in the boiler, the more likely the water is to prime. Boilers will frequently prime badly under forced draft, when otherwise there would be little trouble.

Priming may be detected from the unusual behavior of the water in the gauge-glass, or from the hammering in the steam pipes or cylinder. To avoid a breakdown under such conditions, the speed of the engine should be reduced, the drain-cocks of the cylinders and pipes opened, and the fires eased down. Sometimes, by suddenly shutting the main stop-valve, the pressure in the boiler can be increased sufficiently to overcome the difficulty.

Almost any boiler is likely to prime to some extent; and to obtain as dry steam as possible, several devices are employed. On the top of stationary boilers and locomotives, a *steam dome* is frequently built, from which the steam is drawn, the idea being that less moisture will be found here than if the steam be drawn directly from the main portion of the boiler. In marine work, and sometimes in stationary plants, a *dry-pipe* is used (see Fig. 50). This is merely a large pipe inside the boiler, from which the steam is drawn. The pipe is near the top of the boiler, and the upper side of it is perforated with holes through which the steam may pass. A considerable amount of moisture is in this way prevented from leaving the boiler.

The moisture in steam can be reduced by the familiar process of superheating; but if this, for any reason, is impracticable or undesirable, a steam separator may be used for the purpose of extracting the moisture that comes from the priming of the boiler or from condensation in the steam pipe.

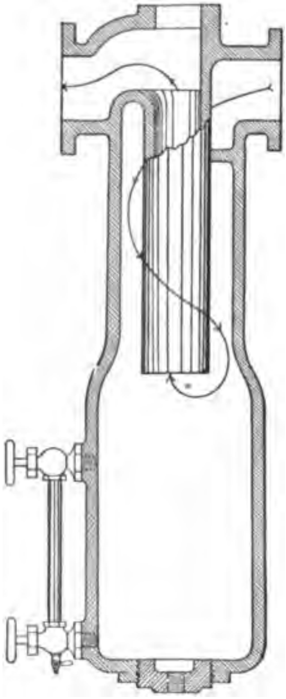


Fig. 59. "Stratton" Separator.

Separators. There are several forms of separator; but all are designed on the general principle that if the direction of the steam current is suddenly changed, or if it is diverted upward and then downward, the water will be separated from the steam and will fall to the bottom of a suitable receptacle. The depth of water collected in the bottom of the separator is readily indicated by a gauge-glass, and it may be drawn off as desired. To prevent the possibility of flooding the separator, it is well to connect it with an automatic trap which will empty it without close attention from the engineer. It is needless, of course, to say that the trap from this separator should be connected to the hot well, and the drip should be returned to the boiler with the loss of as little heat as possible.

In the "Stratton" separator, Fig. 59, the steam enters at one side of a cylinder, flows downward, and then upward through a pipe in the middle. Dry steam escapes from a pipe near the top, on the opposite side from which it enters. The separated water is drained at the bottom.

The "Cochrane" steam separator, shown in section in Fig. 60, is of the baffle-plate type. The branches for the entrance and exit of the steam project from each side of the spherical head. Another branch from the bottom provides for connection with the well. The baffle-plate, which is cast as a part of the head, is ribbed, or corrugated, and has ports at each side for the passage of steam. The area of the ports is large, to prevent loss by friction. A small pipe is inserted in the plate on the outlet side at the bottom of the baffle-plate, to drain

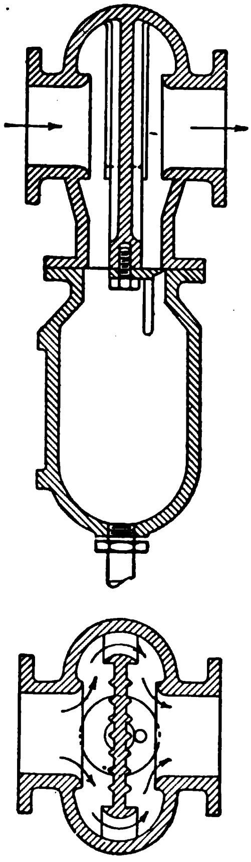


Fig. 60. Sectional Elevation and Plan of "Cochrane" Steam Separator.

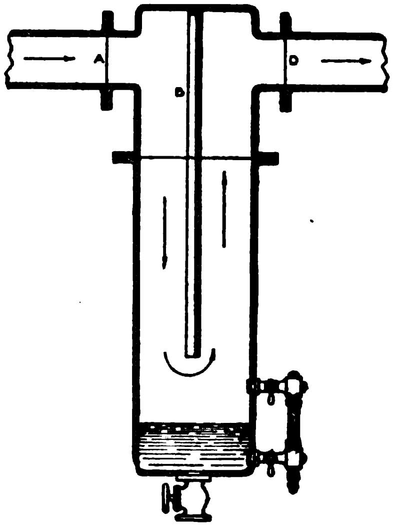


Fig. 61. Separator Designed for Connection to Main Steam Pipe near Engine.

any condensation in the outlet chamber. Steam, entering at the left-hand opening, strikes the baffle-plate and passes to the outlet chamber by means of the two side passages, as shown in the plan, Fig. 60.

A form of separator which is fitted to the main steam pipe near the engine, is shown in Fig 61. Steam enters at *A* and strikes the dash-plate *B*; any water coming with the steam is separated and falls to the bottom. The steam takes the direction indicated by the arrows, and flows out at *D*. This separator is fitted with a gauge-glass which is similar to a boiler gauge-glass.

STEAM TRAPS

Steam traps are used for collecting the water of condensation from steam pipes. They consist of a receptacle with an inlet and outlet valve so arranged that the condensation which collects may flow out, but steam cannot pass.

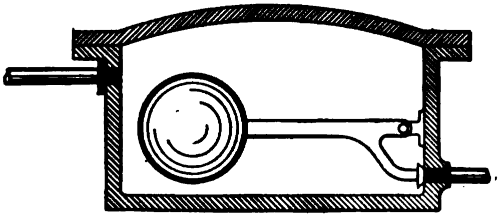


Fig. 62. Simple Steam Trap Operated by Float.

In the *float* trap shown in Fig. 62, the float rises and falls with the change in water-level. When the water-level rises above a certain point, the float opens the discharge valve. The trap shown in Fig. 63 is similar, the float being replaced by a weight *W*, which is nearly counterbalanced by the weight *T*. The raising of *W* by the water opens the valve *V*.

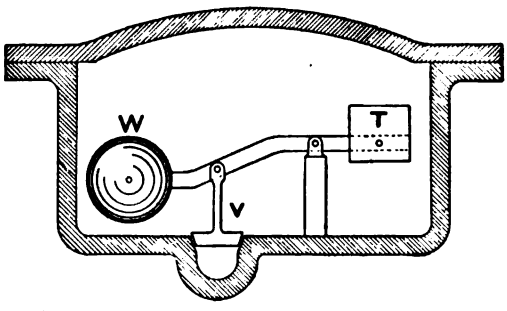


Fig. 63. Steam Trap Operated by Nearly Counter-balanced Weight.

There are other forms called *bucket* traps. In the one shown in Fig. 64, the water enters at *W*. While there is only a little water around the bucket *F*, it floats, and the valve *V* is closed; but when the water rises high enough to flow over the edge, the weight of water in the bucket causes it to sink, and opens the valve *V*. Water is forced up the passage *M*, and out through the pipe *N*, by the pressure of the steam on the surface of the water surrounding the bucket.

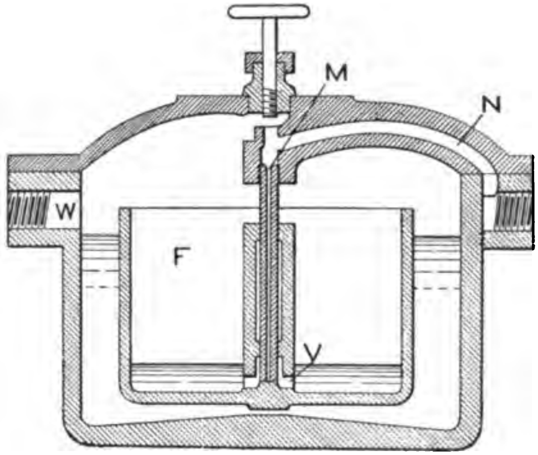


Fig. 64. Bucket Type of Steam Trap.

Another form of trap, called the *differential* steam trap, depends upon a head of water acting on a flexible diaphragm. Water enters at either top or bottom by the pipes *E*, Fig 65. When the water-level rises, it fills the chamber *G* and the pipe *N*. This causes a pressure on the under side of the diaphragm greater than that caused by the spring *H*, which spring acts on the upper side of the diaphragm and tends to keep the valve open. While the pressure below the diaphragm preponderates, the valve *P* remains closed. When the water rises and fills the chamber *J* so as to flow down the pipe *M*, the water-pressure on the upper and lower side of the diaphragm will become equal, because the head of water in *M* is practically the same as that in *N*. The spring will now open the valve *P*, and water will be discharged from the pipe *I*. When the head in *M* falls, the pressure on the under side of the diaphragm again becomes greater, and the valve accordingly closes.

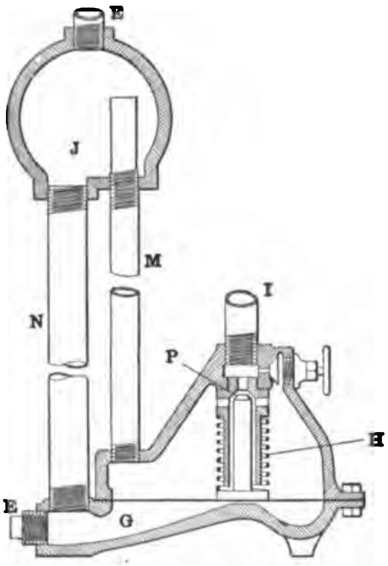


Fig. 65. Differential Steam Trap, Operated by Water-Pressure on a Flexible Diaphragm.

Return Traps. Traps that are used for returning water of condensation to the boiler are called return traps. There are a variety of forms, but the principle of action in all is similar, and is shown in Fig. 66. *B* represents the boiler, and *T* the trap, which is placed a few feet above the boiler. The trap is supplied with steam from the boiler. It is also connected with the boiler by the pipe *P*, in which is a check-valve at *C*. Water of condensation enters the trap through the pipe *E*, in which is a check-valve *H*, until it reaches a depth sufficient to raise the float *F*, which opens the balanced steam valve *V*, called an *equalizing valve*. Steam from the boiler then enters the trap and equalizes the pressure. Since the pressures are equal, water in the trap, because of its height above the water-level of the boiler, will flow to the boiler until the level in the pipe *P* is nearly the same as the water-level in the boiler. As the water-level in the trap falls, the float *F* drops, and the equalizing valve is closed.

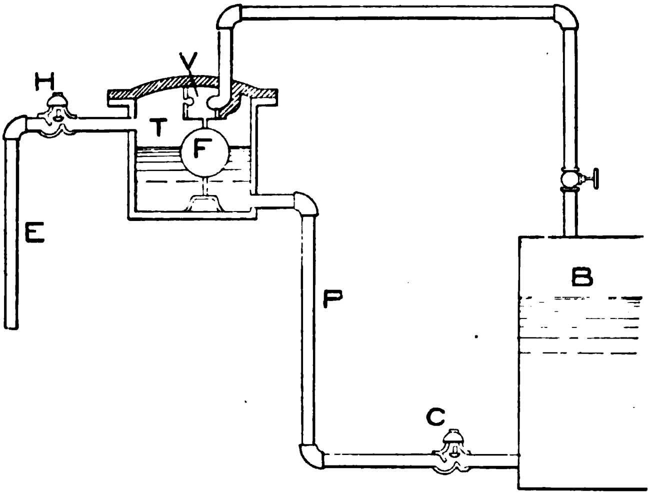


Fig. 66. Diagram Illustrating Operation of Return Trap.

In some forms of return traps, buckets are used instead of floats.

CALORIMETERS

Steam from a boiler is generally accompanied with more or less moisture. This, being mechanically suspended in the steam, cannot readily be measured without the use of special apparatus. An instrument by means of which the percentage of moisture in steam can be determined, is generally called a *calorimeter.* There are several different types of this instrument, only three of which will be described.

The Barrel Calorimeter. This was invented by the distinguished engineer, Mr. G. A. Hirn, and is not only one of the earliest of these devices, but is by all means the simplest and most inexpensive form of calorimeter in practical use. It is shown in Fig. 67. The essential apparatus consists of a barrel holding about 400 lbs. of water, scales for weighing—and nothing more. A pipe with suitable connections leading from the boiler or steam main, conveys the sample of steam to be tested. This pipe should be provided with a valve, and on the end should be a piece of rubber hose which can readily be inserted in the barrel or removed. The principle of this calorimeter is extremely simple. As steam flows through the pipe, it is condensed by the water in the barrel, and the increase in the weight of the barrel after the test indicates the total amount of moist steam condensed, while the rise in temperature of the water in the barrel is an exact measure of the quantity of heat obtained from this moist steam.

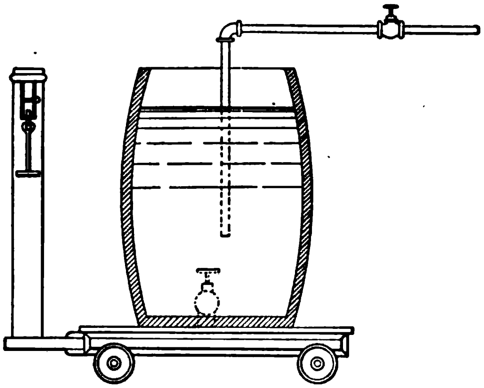


Fig. 67. Details of Barrel Calorimeter.

The steam tables give the number of B. T. U. in dry steam and hot water at various temperatures and pressures; and with this data and the above-mentioned observations made in the barrel, the percentage of steam and moisture can readily be determined.

The sampling pipe usually projects into the steam main a few inches, the end being perforated so that the sample will be drawn from a point near the middle of the pipe. An agitator should be placed in the barrel, so that the water may be thoroughly stirred and a uniform temperature maintained during the test.

To test a sample of steam by this method, fill the barrel about two-thirds full of cold water; place it on platform scales, and carefully note its weight and temperature. The weight of the barrel and fittings, when empty, should of course be known, so that the weight of the water alone can be determined. With the hose removed from the barrel, allow steam to blow through the pipe until it has become thoroughly heated If the sampling pipe is long, it should be wrapped with hair felt or some form of lagging, to prevent condensation during the test. As soon as the pipe line has become thoroughly heated, plunge the hose into the barrel and allow the steam to blow through the water until it has become well heated. Shut off the steam, and carefully note the weight and temperature.

Suppose W = Final weight of water in barrel;
w = Weight of cold, condensing water before steam is turned on;
t_1 = Temperature of the cold water;
t_2 = Temperature of the hot water;
P = Absolute pressure of steam in steam pipe (gauge pressure + atmospheric pressure).

From the steam tables in the back of the book may be found:

q, the B. T. U. in one pound of the liquid contents of the moist steam;
q_1, the B. T. U. in one pound of the cooling water, before the steam was added;
q_2, the B. T U. in one pound of this water after the steam has been added;
r, the heat of vaporization corresponding to the absolute pressure—*i. e.*, B. T. U. given up by one pound of steam condensed into water.

If x equals the percentage of dry steam contained in the supply pipe, $1 - x$ will represent the amount of priming.

$x (W - w)$ = the total amount of dry steam condensed;

$(1 - x) (W - w)$ = the total amount of moisture brought into the barrel by the moist steam.

If q_1 equals the heat in one pound of cooling water, then $q_1 w$ will equal the total heat in the barrel at the beginning.

For the same reason $q_2 W$ will equal the total heat after the steam has been condensed, and $q_2 W - q_1 w$ will equal the total amount of heat gained by the water in the barrel.

If r is the heat of vaporization, then $r\ x\ (W - w)$ will equal the B. T. U. contained in the dry steam; and if q is the heat of the liquid corresponding to the same pressure, then $q\ (1 - x)\ (W - w)$ will equal the B. T. U. contained in the moisture brought over by the steam. It is apparent that the sum of these two quantities will be the total number of B. T. U. brought from the steam main to the water barrel, and must be equal to $q_2 W - q_1 w$, the heat gained by the water in the barrel. The solution of this equation will result in a formula which will save some mathematical computations.

That the method may be perfectly clear, let us first consider a numerical example in full.

Suppose $w = 455$ lbs.
$W = 495$ lbs.
$t_1 = 50°$ F.
$t_2 = 140°$ F.
$P = 75$ lbs.
q (from steam tables) $= 276.9$
q_1 " " " $= 18.1$
q_2 " " " $= 108.2$

Then the total heat in the barrel after condensation, is equal to $(495 \times 108.2) = 53{,}559$ B. T. U.

The total heat before condensation was equal to $455 \times 18.1 = 8{,}235$ B. T. U. Therefore the heat brought over by the moist steam will be $53{,}559 - 8{,}235 = 45{,}324$ B. T. U.

Now, from the steam tables

$$q = 276.9; \text{ and } r = 898.8.$$

The heat given up by condensation of the dry steam will then be $898.8 \times (495-455)x = 40x \times 898.8 = 35{,}952x$; and the heat of the liquid in the moisture and condensed steam will be $40 \times 276.9 = 11{,}076$, making the total heat in the moist steam $= 11{,}076 + 35{,}952x$.

Therefore, $11{,}076 + 35{,}952x = 45{,}324$

$$35{,}952x = 34{,}248$$

$$x = 0.952$$

That is, every pound of moist steam contains .952 lb. dry steam and .048 lb. moisture; or we may say there was 4.8 per cent of priming.

The formula may be derived by the following algebraic work:

Total heat in bbl. after condensation $= W\,q_2$;
Total heat in bbl. before condensation $= w\,q_1$;
Total heat brought over by steam $= W\,q_2 - w\,q_1$;
Heat of liquid in condensed steam $= (W - w)\,q$;
Latent heat in dry steam $= x\,(W - w)\,r$;
Total heat in moist steam $= x\,(W - w)\,r + (W - w)\,q$.

Therefore,

$$x\,(W - w)\,r + (W - w)\,q = W\,q_2 - w\,q_1;$$

$$xr\,(W - w) = W\,q_2 - w\,q_1 - W\,q + w\,q;$$

or, transposing to a more convenient form,

$$x = \frac{w(q - q_1) - W(q - q_2)}{r(W - w)}.$$

The use of this form of apparatus is not especially to be commended, for it is liable to error, and a slight discrepancy in the weights or the temperatures may cause a large error in the result. In the above calculations, no allowance is made for loss of heat through radiation.

Separator Calorimeter. This instrument shown in Fig. 68, consists of a chamber *A*, into which is led a steam pipe *D*, bringing a sample of steam from the boiler or steam main. This pipe leads into an enlargement perforated with small holes, or into a chamber *A* as shown in Fig. 68. The calorimeter separates the moisture from the steam just as a steam separator does; and the exhaust, which is dry steam, passes out of the pipe *P*, wherein is inserted a diaphragm containing small orifices, by means of which the quantity of steam flowing out can be calculated by thermodynamic methods. The exhaust steam can, of course, be led to some form of condensing apparatus, and the condensation weighed, if desired.

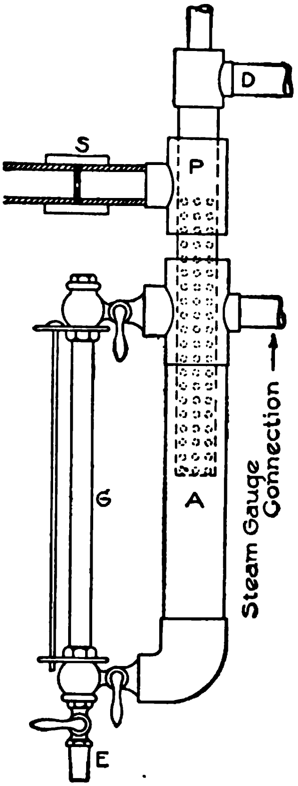


Fig. 68. Separator Calorimeter.

As the steam enters the calorimeter, the moisture is drawn toward the bottom of the chamber. The amount of water collected can readily be read from the gauge-glass at the side, to which a graduated scale should be attached.

The amount of moisture contained in the steam can be weighed directly by drawing it out of the gauge-cock *E*. The amount of dry steam is measured by its flow through the orifices, or by condensation.* If W = weight of steam discharged from the calorimeter, and w = weight of water collected, then the percentage of priming will be $\frac{w}{W + w}$.

---

*NOTE: For principles governing flow of steam through an orifice, consult any treatise on Thermodynamics.

If only a small quantity of steam is used, an allowance must be made for condensation; but if the instrument is well lagged with hair felt or other suitable material, and a sufficient quantity of steam is used, the error from radiation may be neglected. Steam should be allowed to flow through the instrument until it has become thoroughly heated, before beginning the test.

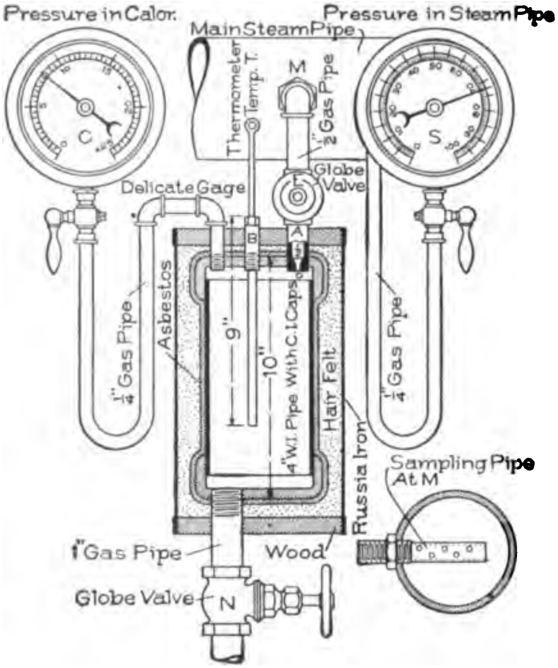


Fig. 69. General Arrangement of Throttling Calorimeter.

Throttling Calorimeter. This was invented by Prof. Cecil H. Peabody, and is made with varying constructive details. Fig. 69 shows the general arrangement. The mixture of steam and water from the boiler is taken from the main steam pipe through what is termed a *sampling* pipe. Various forms of this pipe are made; one arrangement consists of a pipe closed at its inner end, but having numerous holes ¼ inch in diameter drilled staggering around the sides. The calorimeter should be placed as close as possible to the main steam pipe; and the gauge for indicating the pressure in the main steam pipe should be placed on the latter and near the calorimeter. The gauge is sometimes connected to a tee on the pipe leading to the calorimeter; but it is better to have this gauge where the velocity of the flowing steam is less. A valve is placed in the pipe to the calorimeter, below which is inserted a nipple *A* having a small converging orifice D, about two-tenths of an inch in diameter and very carefully made. The object of such an orifice is to determine the weight of steam flowing through the calorimeter, so that an allowance can be made for the loss when testing an engine or boiler, where the net weight used is required. A cup B is screwed into the top, for holding an accurate thermometer. The cup is made of brass, and is filled with oil; but if mercury is used, the cup must be of iron or steel. A delicate gauge C, for determining the pressure in the calorimeter, and a pipe and valves at the bottom, complete the apparatus. The valve N is sometimes omitted, and a simple pipe used, as the throttling is best accomplished by use of the valve E or orifice D. All pipes leading to the calorimeter should be well covered with a good non-conductor.

To use the instrument, proceed as follows: Open wide valves E and N, to bring the apparatus to a uniform temperature; then gradually close E until the steam in the calorimeter is superheated; that is, until the temperature as shown by the thermometer is greater than that corresponding to the absolute pressure determined from the reading of the gauge C and barometric pressure. The result may now be calculated as follows:

x = Weight of steam contained in one pound of the mixture from the main steam pipe or other source;

λ_c = Total heat corresponding to the absolute pressure determined from the reading of the gauge C and barometric pressure; *

T = Temperature as shown by the thermometer;

t_c = Temperature of steam corresponding to the absolute pressure as determined by the reading of the gauge C and barometric pressure;

q_s = Heat of the liquid corresponding to the absolute pressure in the steam pipe;

r_s = Heat of evaporation corresponding to the absolute pressure in the steam pipe;

0.48 = Heat required to superheat the steam one degree Fahrenheit under constant pressure.

Total heat in 1 lb. superheated steam in calorimeter = $\lambda_c + 0.48\ (T - t_c)$ B. T. U.

Total heat in 1 lb. moist steam in steam main = $xr_s + q_s$ B.T.U.

These two quantities are equal; and x, being the only unknown quantity, the equation can easily be solved.

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*NOTE: Some steam tables use H instead of the Greek letter λ (lambda).

$$x = \frac{\lambda_c + 0.48\,(T - t_c) - q_s}{r_s}.$$

Example. Barometric pressure, 14.78 lbs. Absolute pressure in main steam pipe, 87.78 lbs. Absolute pressure in calorimeter, 23.03 lbs. Temperature (T) = 260° F. Then,

$\lambda_c = 1{,}153.68$ $\qquad q_s = 288.1$

$t_c = 235.28$ $\qquad r_s = 890.88$

$$x = \frac{1{,}153.68 + .48\,(260 - 235.28) - 288.1}{890.88} = 0.984 \text{ pound.}$$

Or, in other words, 98.4 per cent of the mixture is steam; or the moisture = 1 — 0.984 = 0.016, or 1.6 per cent.

This form of calorimeter is suitable only for cases where the moisture does not exceed three per cent of the mixture. Its principle is based upon the assumption that there is no loss of heat, in which case steam mixed with a small amount of water is superheated when the pressure is reduced by throttling.

PIPING

Although piping can hardly be considered a boiler accessory, a few general remarks will not be out of place.

Pipes must not only be of sufficient size and strength, but should be so installed as to make ample provision for expansion due to the high temperature when they are filled with steam. The supports for long pipe lines should be arranged somewhat as shown in Fig. 70, which allows the pipe a considerable amount of lateral motion.

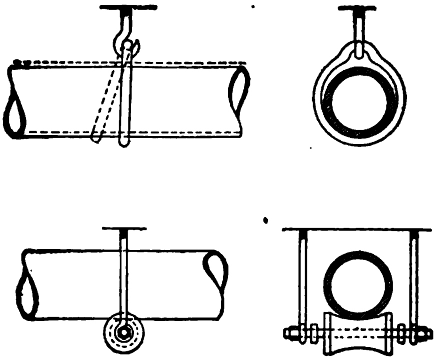


Fig. 70. Side and Transverse Sectional Views Showing Methods of Arranging Supports for Long Pipe Lines.

If the pipe line is long, an *expansion joint* must be provided. Sometimes a curved U-bend may be inserted in the pipe line, which of itself will have flexibility enough to provide for reasonable expansion. Or, if the steam main is not all in one line, a similar bend may be provided, with elbows and nipples, as shown in Fig. 71. In this case, any expansion of the steam main will cause the nipples to turn slightly in the elbows. This motion, of course, is slight, but it is sufficient to prevent rupture. U-bends and swivel-joints are hardly practicable in large pipe; and in such cases a *slip-joint*, made tight by a stuffing gland, is usually provided. If this is done, great care must be taken that the steam main is straight and in perfect alignment, as the pipe may otherwise bind in the expansion joint and cause much damage from leakage.

In marine work, especial care must be taken that the pipe lines are not so rigidly connected together that they will be injured by the working of the ship. This can readily be provided for by laying the pipe in such a way as to provide a simple form of swivel-joint.

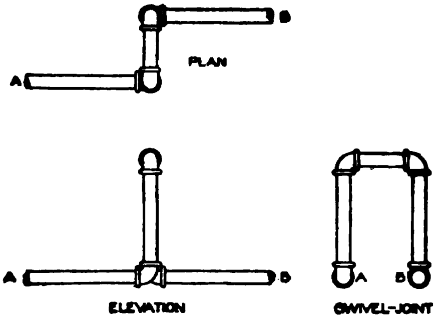


Fig 71. Method of Forming Swivel-Joint in Steam Piping to Counteract Effects of Expansion and Contraction.

The pipe lines should be as straight as possible, to prevent unnecessary friction of the steam and unnecessary condensation; and they should, if possible, be so installed as to leave no pockets wherein condensation may collect. If such a pocket is unavoidable, a drain must be provided, leading from the pocket to the steam trap, whence the condensation may be discharged into the hot well or filter-box, because the collection of water in steam pipes is a source of inconvenience and danger.

The pipe lines should be installed with sufficient slope, so that the condensation will readily drain to a convenient point whence it may be drawn off. This slope should be in the direction of the flow of the steam, as the water will not readily flow otherwise. Great care should be taken that the pipe lines nowhere sag, as such a depression will collect condensation. This may cause very little disturbance unless the pressure of the steam is suddenly raised, in which case the water is liable to flow bodily along the pipe; and if it does not enter the cylinder of the engine and cause damage there, it will cause a serious water-hammer which may rupture the elbows of the pipe and may endanger life.

Formerly, when low pressures were used, cast iron was a common material for a main steam pipe leading from the boiler to the engine, but the higher pressures of to-day require the best wrought iron or steel. In marine work, copper is commonly used; but with the advent of higher and higher pressures, copper fails to give the requisite strength, and it has to be reinforced with wire or iron bands. At pressures not over 150 lbs., copper pipes may be used, by the British Board of Trade rules, 15 inches in diameter; but at 200 lbs., copper pipes are not allowed over 10 inches in diameter. For large sizes, riveted iron or steel pipe may be used. For high pressures, cast-steel fittings are required by the U. S. Steamboat Inspection rules. There was always danger that the large copper pipe would burst; and it is now the common practice to use steel for such purposes.

Large steam pipe is made in sections which can be riveted together. The small sizes are fitted with the ordinary type of flange, and the sections may be bolted together, a suitable gasket being used between the two flanges to make a steam-tight joint. The flanges are machined perfectly smooth, and the packing may consist of rubber and fiber reinforced with wire insertion, or of asbestos, or of corrugated copper.

The true inside diameter of steam, gas, or water pipe is not always the same as the size of the pipe as popularly known. For instance, what is called "3-inch" pipe has an actual inside diameter of 3.067 inches, and 3.5 inches outside diameter. The actual sizes of pipe, inside and outside, can be found in any handbook or steamfitter's catalogue.

LAGGING

When steam pipes are exposed to the air, a considerable amount of condensation will collect in them, depending on the condition of the surface of the pipe, on the difference in temperature between the steam and the surrounding air, and on the velocity of the steam through the pipe. This condensation will cause a large amount of heat to be lost to useful work, and will make the dangers of water-hammer possible unless carefully drained. Tests have shown that about 2 B. T. U. are lost per square foot of pipe per hour per degree of difference in temperature. While the loss for a few hours is not likely to be great, yet, if taken for an entire year throughout a considerable length of pipe, the sum total will be very large indeed. The following table gives some idea of the loss of heat through bare pipe at 200 lbs. pressure:

HEAT LOSSES IN BARE PIPES

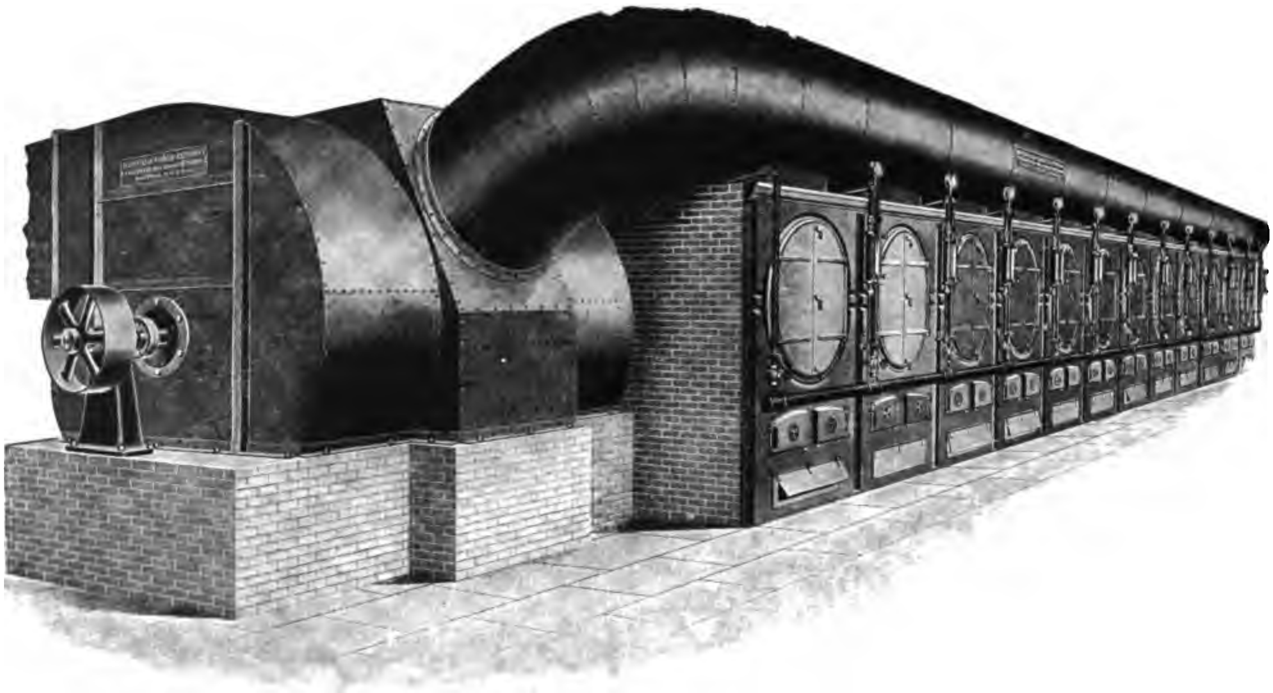
CONDITION OF PIPE	B. T. U. LOSS PER SQ. FT. PER MINUTE
New Pipe	11.96
Painted Glossy Black	12.10
Painted Glossy White	12.02
Fair Condition	13.84
Rusty	14.20
Coated with Cylinder Oil	13.90
Painted Dull Black	14.40

VARIATION OF HEAT LOSS WITH PRESSURE

PRESSURE	HEAT LOSS B. T. U. PER SQ. FT. PER MINUTE
340	15.97
200	13.84
100	8.92
80	8.04
60	7.00
40	5.74

A full account of some interesting tests can be found in a paper entitled *Protection of Steam-Heating Surfaces*, by C. L. Norton, Vol. XIX, Proceedings of the American Society of Mechanical Engineers, 1898, from which these tables have been taken.

Pipe Coverings. To make this loss from radiation as small as possible, it is customary to cover the pipe or boiler with some material which will prevent loss of heat and which will not burn. There is considerable difference in the value of various substances as preventatives of heat radiation. Their value varies nearly in an inverse ratio to their conducting power; but due allowance must be made for the possible deterioration of the pipe covering. The following table gives the relative value of various substances with reference to their ability to prevent radiation of heat. For purposes of comparison, the value of wool is taken as the standard:



BOILERS EQUIPPED WITH DUPLEX THREE-QUARTER HOUSING FANS.
Induced Draft Plant at Watkins Salt Company.
Buffalo Forge Company.

RELATIVE VALUES OF VARIOUS PREVENTATIVES OF RADIATION OF HEAT

Felt, Hair, or Wool	100
Asbestos Sponge	98
Air-Cell Asbestos	89
Mineral Wool	68 – 83
Carbonate of Magnesia	67 – 76
Charcoal	63
Sawdust	61 – 68
Asbestos Paper	47
Wood	40 – 55
Asbestos, Fibrous	36
Plaster of Paris	34
Air Space (Undivided)	22

There are many patented coverings which are very efficient, but they are too numerous even to mention. The above-mentioned article from the Proceedings of the American Society of Mechanical Engineers gives the results of tests of several of these coverings. A good protection is afforded by air confined in minute cells, such as is to be had in the air-cell asbestos board; this is made by cementing together several layers of asbestos paper which have been corrugated or indented by machinery so as to form minute air-cells. The more minute the subdivision of these cells, the better the protection is likely to be. Hair felt is one of the most efficient non-conductors, because it is very porous and contains a large number of air-cells. It is not one of the best coverings, however, because it is liable to deteriorate, and its life on high-pressure pipes is not likely to be more than four or five years. On low-pressure work it may last for a considerably longer time.

Mineral wool, a fibrous material made from blast-furnace slag, is an efficient and noncombustible covering, but is brittle and liable to fall off.

The coverings most easily applied to pipes are those applied in sectional form, which clasp around the pipe and are fastened by brass bands at convenient intervals. Such coverings are made both of asbestos and of magnesia, and are usually of about 1 inch in thickness.

A good, cheap covering can be made by wrapping several layers of asbestos paper around the pipe, and then covering these layers with a layer of hair felt perhaps $\frac{3}{4}$ inch thick, the whole being wrapped in canvas. On low-pressure steam pipes this covering will last ten to fifteen years.

Cork is perhaps one of the most satisfactory coverings from the point of radiation loss, but is rather more expensive than asbestos or magnesia.

It has generally been the impression that it is not economical to cover a pipe to more than one inch in thickness. This will depend upon the cost of the covering and the length of time it is likely to last. If it does not last more than five years, one inch is probably the most economical thickness; but if the life of the covering is likely to be ten years or more, a second inch in thickness can be applied to advantage. For instance, in the above-mentioned tests, in the case of "Nonpareil cork," increasing the thickness from one to two inches raised the cost from $25 to $30 per 100 square feet, and increased the net saving in five years by $10, and by $30 in ten years. A third inch of covering did not produce saving enough to pay for its cost. In each case with the asbestos fire-board, a second inch in thickness showed a saving of $20 in ten years, while the third inch in thickness showed an actual loss from the dollars-and-cents point of view. It would be well to remark that it is of great importance that the pipe covering should be kept in repair, for a loose-fitting covering is of little value.

Boiler Coverings. Much the same remarks may be made with regard to boiler covering as have been made with regard to pipe covering, except that the covering put on boilers is usually somewhat less efficient and is applied in greater thickness. Probably one of the best coverings for a marine boiler—or, in fact, for any internally-fired boiler—is a layer of air-cell asbestos board, covered with a coating perhaps two inches thick of magnesia or asbestos. This comes in powder form, and when mixed with water can be readily applied with a trowel. Coverings on boilers are best placed directly against the shell without an air-space, so that any leak in a joint or rivet will reveal the spot by moistening the covering; otherwise the escaping water may run down through the air-space and appear at some remote point, the leak thus being difficult to locate.

An efficient covering for boilers is made of either magnesia or asbestos in the form of blocks of the proper curvature, which can lie directly against the boiler; but this form of covering is rather more expensive than the asbestos or magnesia cement. To secure an extra hard finish a coating of plaster of Paris may be put on outside the magnesia or asbestos. No boiler or pipe covering should contain sulphate of lime, as this is liable to cause corrosion.

If an internally-fired boiler is properly lagged, there is little danger that any large amount of heat will be lost, as the heat of the fire must pass through the water before radiating. This is not true with an externally-fired boiler, where a considerable amount of heat may radiate through the brick setting of the boiler without coming in contact with the boiler at all. The setting of such a boiler should be arranged with properly confined air-spaces; and an efficient protection from the radiation of heat at the top of the boiler may be had by allowing a slight space between the boiler and the top covering for the circulation of the hot gases of combustion. These are on their way to the chimney; and as they are necessarily hotter than the water in the boiler, they prevent radiation at this point.

HORSE-POWER OF BOILERS

The unit which we call the horse-power is *arbitrary.* Assuming that 30 pounds of steam are required per horse-power per hour for an average engine, this unit for boilers has been adopted.

One (1) horse-power is the evaporation of 30 pounds of water per hour, from a temperature of 100° F. into steam at 70 pounds gauge pressure. This is considered equivalent to the evaporation of 34½ pounds per hour from and at 212° F. A boiler horse-power is equivalent to 33,327 B. T. U. per hour.

As all boilers do not generate steam at the same pressure and from the same temperature of feed-water, it is necessary to reduce the *actual* evaporation to an *equivalent* evaporation. Unless this is done, the relative performances of boilers cannot be compared.

For this comparison, the actual evaporation is reduced to the equivalent evaporation from and at 212° F. That is, we suppose the water to be fed at 212° and evaporated into steam at 212°.

Let W = Water actually evaporated in pounds;

H = Total heat of steam above 32° F., at actual absolute pressure;

T = Temperature of feed water;

w = Equivalent evaporation from and at 212° F.

Since 966 B. T. U. are necessary to evaporate one pound of water

FACTORS OF EVAPORATION

Temperature Feed-Water. Fahrenheit.	PRESSURE IN POUNDS PER SQUARE INCH ABOVE THE ATMOSPHERE (GAUGE PRESSURE.)																					Temperature Feed-Water. Fahrenheit.
Degr's	0	5	15	25	35	45	55	65	75	85	95	105	115	125	135	145	155	165	175	185	200	Degr's
32	1.187	1.192	1.199	1.204	1.209	1.212	1.216	1.218	1.221	1.223	1.226	1.228	1.230	1.231	1.233	1.235	1.236	1.238	1.239	1.240	1.241	32
35	1.184	1.189	1.196	1.201	1.206	1.209	1.213	1.215	1.218	1.220	1.223	1.225	1.227	1.228	1.230	1.232	1.233	1.235	1.236	1.237	1.238	35
40	1.179	1.184	1.191	1.196	1.201	1.204	1.208	1.219	1.213	1.215	1.218	1.220	1.222	1.223	1.225	1.227	1.228	1.230	1.231	1.232	1.233	40
45	1.173	1.178	1.185	1.190	1.195	1.198	1.202	1.204	1.207	1.209	1.212	1.214	1.216	1.217	1.219	1.221	1.222	1.224	1.225	1.226	1.227	45
50	1.168	1.173	1.180	1.185	1.190	1.193	1.197	1.199	1.202	1.204	1.207	1.209	1.211	1.212	1.214	1.216	1.217	1.219	1.220	1.221	1.222	50
55	1.163	1.168	1.175	1.180	1.185	1.188	1.192	1.194	1.197	1.199	1.202	1 204	1.206	1.207	1.209	1.211	1.212	1.214	1.215	1.216	1.217	55
60	1.158	1.163	1.170	1.175	1.180	1.183	1.187	1.189	1.192	1.194	1.197	1.199	1.201	1.202	1.204	1.206	1.207	1.209	1.210	1.211	1.212	60
65	1.153	1.158	1.165	1.170	1.175	1.178	1.182	1.184	1.187	1.189	1.192	1.194	1.196	1.197	1.199	1.201	1.202	1.204	1.205	1.206	1.207	65
70	1.148	1.153	1.160	1.165	1.170	1.173	1.177	1.179	1.182	1.184	1.187	1.189	1.191	1.192	1.194	1.196	1.197	1.199	1.200	1.201	1.202	70
75	1.143	1.148	1.155	1.160	1.165	1.168	1.172	1.174	1.177	1.179	1.182	1.184	1.186	1.187	1.189	1.191	1.192	1.194	1.195	1.196	1.197	75
80	1.137	1.142	1.149	1.154	1.159	1.162	1.166	1.168	1.171	1.173	1.176	1.178	1.180	1.181	1.183	1.185	1.186	1.188	1.189	1.190	1.191	80
85	1.132	1.137	1.144	1.149	1.154	1.157	1.161	1.163	1.166	1.168	1.171	1.173	1.175	1.176	1.178	1.180	1.181	1.183	1.184	1.185	1.186	85
90	1.127	1.132	1.139	1.144	1.149	1.152	1.156	1.158	1.161	1.163	1.166	1.168	1.170	1.171	1.173	1.175	1.176	1.178	1.179	1.180	1.181	90
95	1.122	1.127	1.134	1.139	1.144	1.147	1.151	1.153	1.156	1.158	1.161	1.163	1.165	1.166	1.168	1.170	1.171	1.173	1.174	1.175	1.176	95
100	1.117	1.122	1.129	1.134	1.139	1.142	1.146	1.148	1.151	1.153	1.156	1.158	1.160	1.161	1.163	1.165	1.166	1.168	1.169	1.170	1.171	100
105	1.111	1.116	1.123	1.128	1.133	1.136	1.140	1.142	1.145	1.147	1.150	1.152	1.154	1.155	1.157	1.159	1.160	1.162	1.163	1.164	1.165	105
110	1.106	1.111	1.118	1.123	1.128	1.131	1.135	1.137	1.140	1.142	1.145	1.147	1.149	1.150	1.152	1.154	1.155	1.157	1.158	1.159	1.160	110
115	1.101	1.106	1.113	1.118	1.123	1.126	1.130	1.132	1.135	1.137	1.140	1.142	1.144	1.145	1.147	1.149	1.150	1.152	1.153	1.154	1.155	115
120	1.096	1.101	1.108	1.113	1.118	1.121	1.125	1.127	1.130	1.132	1.135	1.137	1.139	1.140	1.142	1.144	1.145	1.147	1.148	1.149	1.150	120
125	1.091	1.096	1.103	1.108	1.113	1.116	1.120	1.122	1.125	1.127	1.130	1.132	1.134	1.135	1.137	1.139	1.140	1.142	1.143	1.144	1.145	125
130	1.085	1.090	1.097	1.102	1.107	1.110	1.114	1.116	1.119	1.121	1.124	1.126	1.128	1.129	1.131	1.133	1.134	1.136	1.137	1.138	1.139	130
135	1.080	1.085	1.092	1.097	1.102	1.105	1.109	1.111	1.114	1.116	1.119	1.121	1.123	1.124	1.126	1.128	1.129	1.131	1.132	1.133	1.134	135
140	1.075	1.080	1.087	1.092	1.097	1.100	1.104	1.106	1.109	1.111	1.114	1.116	1.118	1.119	1.121	1.123	1.124	1.126	1.127	1.128	1.129	140
145	1.070	1.075	1.082	1.087	1.092	1.095	1.099	1.101	1.104	1.106	1.109	1.111	1.113	1.114	1.116	1.118	1.119	1.121	1.122	1.123	1.124	145
150	1.065	1.070	1.077	1.082	1.087	1.090	1.094	1.096	1.099	1.101	1.104	1.106	1.108	1.109	1.111	1.113	1.114	1.116	1.117	1.118	1.119	150
155	1.059	1.064	1.071	1.076	1.081	1.084	1.088	1.090	1.094	1.095	1.098	1.100	1.102	1.103	1.105	1.107	1.108	1.110	1.111	1.112	1.113	155
160	1.054	1.059	1.066	1.071	1.076	1.079	1.083	1.085	1.088	1.090	1.093	1.095	1.097	1.098	1.100	1.102	1.103	1.105	1.106	1.107	1.108	160
165	1.049	1.054	1.061	1.066	1.071	1.074	1.078	1.080	1.083	1.085	1.088	1.090	1.092	1.093	1.095	1.097	1.098	1.100	1.101	1.102	1.103	165
170	1.044	1.049	1.056	1.061	1.066	1.069	1.073	1.075	1.078	1.080	1.083	1.085	1.087	1.088	1.090	1.092	1.093	1.095	1.096	1.097	1.098	170
175	1.039	1.044	1.051	1.056	1.061	1.064	1.068	1.070	1.073	1.075	1.078	1.080	1.082	1.083	1.085	1.087	1.088	1.090	1.091	1.092	1.093	175
180	1.033	1.038	1.045	1.050	1.055	1.058	1.062	1.064	1.067	1.069	1.072	1.074	1.076	1.077	1.079	1.081	1.082	1.084	1.085	1.086	1.087	180
185	1.028	1.033	1.040	1.045	1.050	1.053	1.057	1.059	1.062	1.064	1.067	1.069	1.071	1.073	1.074	1.076	1.077	1.079	1.080	1.081	1.082	185
190	1.023	1.028	1.035	1.040	1.045	1.048	1.052	1.054	1.057	1.059	1.062	1.064	1.066	1.067	1.069	1.071	1.072	1.074	1.075	1.076	1.077	190
195	1.018	1.023	1.030	1.035	1.040	1.043	1.047	1.049	1.052	1.054	1.057	1.059	1.061	1.062	1.064	1.066	1.066	1.069	1.070	1.071	1.072	195
200	1.013	1.018	1.025	1.030	1.035	1.038	1.042	1.044	1.047	1.049	1.052	1.054	1.056	1.057	1.059	1.061	1.062	1.064	1.065	1.066	1.067	200
205	1.001	1.012	1.019	1.024	1.029	1.032	1.036	1.038	1.041	1.043	1.046	1.048	1.050	1.051	1.053	1.055	1.056	1.058	1.059	1.060	1.062	205
210	1.002	1.007	1.014	1.019	1.024	1.027	1.031	1.033	1.036	1.038	1.041	1.043	1.045	1.046	1.048	1.050	1.051	1.053	1.054	1.055	1.057	210
212	1.000	1.005	1.012	1.017	1.022	1.025	1.029	1.031	1.034	1.036	1.039	1.041	1.043	1.044	1.046	1.048	1.049	1.051	1.052	1.053	1.056	212

from and at 212° F., the equivalent evaporation may be found from the formula,

$$W\,(H + 32 - T) = 966w, \text{ or } w = \frac{W\,(H + 32 - T)}{966}.$$

Then the horse-power of the boiler is:

$$\text{H. P.} = \frac{w}{34.5}.$$

The above method is considerably shortened by substituting for the quantity $\frac{H + 32 - T}{966}$, the number found in the accompanying table (page 80) which corresponds to the actual feed-water temperature and steam pressure.

For example, a boiler is required to furnish 2,100 pounds of steam per hour. If the gauge pressure is 85 pounds, and the feed-water enters at 50° F., what is the equivalent evaporation, and what is the horse-power?

From the table, the factor for 85 pounds pressure and 50° F. is 1.204. Then the equivalent evaporation would be $1.204 \times 2{,}100 = 2{,}528.4$ pounds; and $\frac{2{,}528.4}{34.5} = 73$ (approx.) = the H. P.

CORROSION AND INCRUSTATION

There are several causes which tend to shorten and destroy the life of every boiler. These may be divided into two general classes, chemical and mechanical, and are usually the result of improper feed-water or of improper care. Pure water, free from air and carbon dioxide, has no evil effect on the iron; but all natural waters, whether from rain, lake, river, or sea, contain air and a little carbon dioxide in solution, and such water will cause iron to corrode, even though no other impurities are present.

Sea water, heated under a steam pressure of 30 lbs., even if it contains no air, will liberate a small amount of hydrochloric acid, which instantly attacks the iron of the boiler unless counteracted by some chemical agent.

External Corrosion. There are two forms of corrosion, external and internal. External may be due to faulty setting, to improper care, or to moisture from external sources or from leakage from joints and valves. A large amount of external corrosion is the result of setting boilers in a mass of brickwork, which readily absorbs moisture, and which, when not under fire, is likely to keep the boiler-plates damp. The exterior of a boiler encased in brickwork, moreover, is not so easily accessible, and a considerable amount of deterioration may take place without being readily detected. The leakage from a joint, although slight, may, if long continued, badly corrode the boiler.

Internally-fired boilers are supported on saddles and are easily accessible; and the magnesia or asbestos lagging with which they are usually covered will tend to absorb a certain amount of moisture, which will be given off when hot, thus helping to keep the boiler dry. If a leak occurs of appreciable size, the covering will become softened and its presence will be detected at once, and repairs can be made before any serious damage is done. The exterior of an internally-fired boiler, being at all times accessible, can be properly taken care of, which is not true of a boiler set in brickwork. Rivets and riveted joints should as far as possible be kept out of contact with the fire.

Internal Corrosion. This is the result of the chemical action of impure feed-water. It may occur in the form of a general corrosion or wasting-away of the boiler-plates, or in the form of pitting or grooving, the effects of which are likely to be local. Pitting and general corrosion are entirely the result of chemical action, while grooving is the result of chemical and mechanical action combined.

It is not easy to discover general corrosion, because it acts more or less uniformly over a large surface. Sometimes the rivet-heads rust in proportion to the plates, so that the wasting-away of the plates is not easily noticeable. A uniform corrosion is the hardest to detect, and can usually be discovered only by drilling the boiler and gauging the thickness of the plate. If the thickness of the plate is found to be materially reduced, the working pressure of the boiler should be lowered in proportion.

Sometimes the water will attack the plates only in the vicinity of the water-line, in some instances confining the damage to a belt 6 inches or 8 inches wide. Sometimes a few rivets below water-level will be corroded, the rest remaining in a comparatively good condition. Often the stays are weakened more rapidly than the plates, and the screw-threads of a stay may be badly corroded while the shank of the stay remains uninjured.

Pitting. Fatty acids, which are likely to come over in the feed-water if vegetable oils are used to lubricate the cylinder, are especially active in the production of small pits throughout the interior of the boiler. Pitting appears in the form of small holes or in patches from $\frac{1}{4}$ inch to 1 inch in diameter, or even as irregularly shaped depressions. If the holes are small and close together, the plate is said to be *honeycombed.* It is generally believed that this phenomenon, the result of chemical action, is due to a lack of homogeneity in the material of the boiler, although an entirely satisfactory explanation has not yet been given. Pitting may also be caused by galvanic action, which may take place especially if sea water is used. As pitting occurs when there is no cause whatever for galvanic action, this can be only a secondary cause at best. It is reasonable to suppose that acids will attack the most susceptible portions of the plate; and if there is any lack of homogeneity in the iron, it is probable that the places or spots most favorable to chemical attack will suffer first.

Grooving. Grooving is probably the result of straining, springing, or buckling of the plates, aided by local corrosion or by the same forces which cause pitting. Straining of the plates may be due to insufficient or improper staying, thus causing the plates to spring back and forth as the steam pressure varies. This phenomenon is most commonly found in stationary boilers of the "Cornish" or "Lancashire" types appearing in the flat end-plates around the edge of the angle iron, or in the root of the angle iron. Too rigid staying of the ends by gussets or diagonal stays, or too great a difference in expansion between different parts, is almost sure to produce grooves.

Internal grooving may be caused as the direct result of excessive calking, which, by injuring the surface of the metal, exposes it to the corrosive action of the feed-water. It is to be expected that if strains which cause the plates to come and go are set up in the boiler—especially if the stresses can be concentrated along a definite line—a weakness will be developed there, and it will be a susceptible point for chemical attack. Sometimes grooving is so fine as to appear to be a mere crack. But the crack, although perhaps only $\frac{1}{64}$ inch in width, may extend into the plate for a considerable depth. Grooves are not readily detected, and if allowed to continue for any length of time are likely to produce serious results.

Prevention. The best way to prevent internal corrosion is to use water that has no corrosive effect on the plates. If internal corrosion has begun, a change of feed-water may prolong the life of the boiler, but in many instances it is cheaper to build a new boiler than frequently to change the water supply. Sometimes the introduction of a thicker plate at places where the water is found to be most active will be advisable; but, as these plates are stronger than the rest of the boiler, the strains will not be uniformly distributed, and stresses are likely to concentrate along the edge of this heavy plate, which will be a susceptible point for the formation of grooves.

The acidity of the feed-water may be neutralized by some alkaline substance, such as soda, before it enters the boiler. The amount of soda to be used varies with the acidity of the water; but it should always be used in the smallest possible quantity, as the soda is likely to produce priming in the boiler and will be injurious if there is much salt present. Vegetable oils should not be used for cylinder lubrication if the condensation is to be fed back to the boiler, as such oils contain acids which will always produce injurious effects. Mineral oils alone should be used.

To allow for a general corrosion, $\frac{1}{16}$ inch to $\frac{3}{16}$ inch extra thickness of shell should be provided. All seams of a boiler should be tight, and no welded tubes should be used, as pitting and grooving are likely to occur in the vicinity of the weld. When not in use, no moist air should be allowed in the boiler. A boiler can be thoroughly dried out either by the application of heat or by placing in it lime, which will readily absorb the moisture.

The water fed to the boiler should be thoroughly filtered to remove as much grease as possible, for, although mineral oil is not likely to cause pitting, it has a serious effect in the formation of boiler scale.

Incrustation. The incrustation formed by the accumulation of the deposit of sediment in the feed water, is called *scale* or *sludge.* The solid matter in the feed-water may be precipitated by the rise in temperature, or left behind as the result of the evaporation of the water. These solids, unless blown out, are liable to become hardened on the inner surface of the boiler. A thin coating of scale in itself is beneficial, for it keeps the water from direct contact with the iron, and prevents corrosion and pitting; but the danger is that if a thin scale forms, a thicker one will form, and this heavy scale, being a poor conductor of heat, not only causes considerable waste of fuel, but allows the plates next the furnace to become overheated, with the result that they are likely to give way, and the boiler may collapse.

The amount of solid matter in solution is measured in grains per U. S. gallon. The quantity varies greatly in waters from different sources, but is seldom over 40 grains per gallon. It is not the quantity of matter in solution, but its nature, that determines the influence of feed-water. With proper attention to the boiler, the presence of a certain amount of carbonate or sulphate of soda would not be injurious; while the same number of grains per gallon of salts of lime would cause serious trouble. Salts of lime (calcium), together with carbonate of magnesia, are the solids most frequently found, and are the most troublesome. *Hard water* contains considerable quantities of lime. So-called *soft water* has usually but little solid matter in suspension, but it may contain vegetable or organic impurities that will cause corrosion or pitting.

The oil used in the engine is likely to get into the boiler through the feed-water, if it is not carefully filtered or passed through a *grease-extractor*. The oil is likely to be deposited on the sides and tubes of the boiler, and not only is a poor conductor of heat, but, mingling with the sediment which is precipitated from the hot water, produces a mixture which is readily baked onto the boiler-plates and is especially obstinate and difficult to remove. There are efficient grease-extractors now on the market, which will remove practically every trace of oil.

Carbonate of Lime. Carbonate of lime is held in solution in water by an excess of carbon dioxide. As the water is heated, the excess of carbon dioxide, or carbonic acid, is driven off, and the carbonates will be precipitated in the form of a whitish or grayish sediment of the consistency of mud. If these precipitates are not mixed with impurities, they may be washed out of the boiler after it has been allowed to cool; but if there is oil, organic matter, or sulphate of lime, the deposits are likely to become hard. They may readily be drawn off through the bottom blow-out; but if there is much pressure in the boiler, the blow-out valve should be opened only for a very short time. If a considerable amount of water is blown out while the boiler is still very hot, a large part of this precipitation is likely to be baked onto the tubes and interior of the boiler in a manner that defies removal. Short and frequent blowings will accomplish the desired result; for while the boiler is in action these precipitates are more or less in motion, and frequent blowing will keep the boiler clear. Oil and various organic matters rising to the surface can easily be removed by frequently opening the surface blow-out.

Sulphate of Lime. This troublesome salt, like the carbonate of lime, is precipitated with a rise of temperature; and at 280° F., none is left in solution. This sediment is likely to form a hard, adhering scale; but if a little carbonate of soda, or soda ash, is introduced with the feed water, calcium carbonate is precipitated in the form of a white powder which can be readily washed out. The carbonate of soda should be introduced at regular intervals, a portion of it being dissolved in water which can be mixed with the feed in the hot well. As little soda as possible should be used, as it is likely to cause priming and foaming. The hardness of the scale formed by the sulphate of lime depends on the other impurities in the water and on the temperature; and consequently the amount of soda that can safely be used can be determined only by trial. Ammonium chloride, commonly called *sal-ammoniac*, is sometimes used to break up these lime compounds, but is not always desirable, as it may break up the chlorides if other conditions are right, thus forming free chlorine, which attacks the boiler.

Carbonate of Magnesia is seldom found in such large quantities as calcium salts. Like the carbonate of lime, it is precipitated in hot water. If there is any oil or organic matter present, it is likely to form an injurious precipitation.

Iron Salts form a reddish incrustation which is very injurious to boiler-plates. Brakish water containing chloride of magnesium is also injurious; for, when heated, the chloride decomposes, forming magnesia and hydrochloric acid, the latter rapidly corroding iron.

A piece of thick scale broken from the plates of the boiler, will show a series of layers of various thickness, some of them crystalline and some amorphous. Between these hard layers are frequently found layers of soft or earthy matter.

Nothing definite is known in regard to the loss of heat caused by scale on heating surfaces, for there are too many circumstances to be considered to admit of exact calculation. It has been stated that a layer $\frac{1}{16}$ inch thick in the tubes of multitubular boilers, is equivalent to a loss of from 15 to 20 per cent of fuel. The loss increases rapidly with the thickness of the scale. A uniform coating of scale is not nearly so harmful as irregular deposits, for in the latter case the evil effects of overheating are likely to be produced, and overheating will result where it is least suspected.

Prevention. Incrustation may be prevented by precipitating the scale-forming substances before the feed-water reaches the boiler, by the introduction of chemical compounds to neutralize the evil effects, or by removing the sediment before it becomes hard. Scale may, of course, be removed by hand from the interior of the boiler; but this is a slow and tedious process. One of the chief objections to removing scale by hand is that the surfaces of the boiler are likely to become abraded by the chipping tools, and this offers excellent opportunity for pitting and local corrosion to set in.

Scale has sometimes been removed by blowing the boiler off at comparatively high pressure, and then filling it with cold water. This causes a severe contraction of the plates, and is likely to loosen the scale; but it will at the same time cause serious injury to the boiler, and is a practice that should not be tolerated.

After the impurities are deposited in the boiler, they may be removed by the blow-out apparatus; and if it is possible to "lay off" the boiler occasionally, it should be allowed to cool down slowly, and then the water may be drawn off and the boiler properly washed out. A considerable amount of heat is abstracted from the boiler by frequent blowing-off, and this is a matter of direct loss, but it is nothing like so much as would be caused by the formation of scale.

Water may be purified to a certain extent by passing it through a *purifier* before allowing it to enter the boiler. The carbonate and sulphate of lime are precipitated at the same time that the water is heated. The purifier was referred to under the topic of "Feed-Water Heaters." The use of soda for the neutralization of sulphate of lime has already been spoken of; but various compounds are on the market for overcoming the evil effects of other solids; and it is possible, by an analysis of the feed water, to prescribe a boiler compound that will give satisfactory results. Cheap compounds, sold without reference to the analysis of the feed-water, should be avoided. Caustic soda may be used instead of the carbonate, but should be used in small quantities. A rapid circulation of the water will prevent the formation of scale, the sediment being swept from the tubes or shell into the mud-drum, whence it may be blown off. This is one of the chief advantages claimed for water-tube boilers.

Zinc plates have frequently been used to prevent corrosion and incrustation. The brass fittings are likely to set up a galvanic action with the steel plates; but if the zinc is put in, it will be acted upon instead of the iron, which otherwise might be rapidly wasted. It is claimed that this galvanic action prevents the formation of scale by liberating hydrogen at the exposed surfaces. The zinc neutralizes the free acids by combining with them, and takes the place of iron in causing precipitation of copper salts when present.

Kerosene oil is used to a considerable extent to prevent the formation of scale and to assist in its removal. It breaks up and loosens hard scale, and prevents its formation. About one quart a day is sufficient for each 100 horse-power of the boiler.

BOILER EXPLOSIONS

Safety is one of the first requisites in a steam boiler, and must be assured not only by proper design in the beginning, but by subsequent care and proper maintenance. The evil effects of corrosion and incrustation have been clearly shown; and it is apparent that a boiler which has suffered materially from either cause is not in condition to stand full steam pressure. Since the explosion of a boiler, especially in a city or a factory, is likely to prove fatal to many people and to cause the destruction of considerable property, not only by the explosion itself but also by fire, which almost invariably follows such an occurrence, it is impossible to lay too great emphasis on the necessity of seeing that the boiler is in proper working condition.

All boilers must be carefully tested—land boilers, by the State Inspectors; marine boilers, by the United States Inspectors. The boilers are carefully examined inside and outside, and subjected to a hydraulic pressure test 50 per cent greater than the designed pressure of steam; and if there is the slightest sign of pitting or corrosion, the boiler-plates may be drilled and the thickness calipered, the hole being refilled by a proper plug. If a boiler passes inspection, a subsequent explosion will probably be the result of mismanagement, although inspection is not infallible.

The owner of the boiler is usually held liable in case of explosion; but may protect himself from financial loss by insurance against accident in any of the boiler insurance companies. If so insured, the Insurance Inspector, as well as the State Inspector, examines the boiler; and there is consequently less likelihood of an explosion, for an insurance inspector will naturally be exceedingly careful in the interests of his company.

The damage done by an explosion is due to the energy stored in the hot water, which energy can be calculated by thermodynamic methods. If a boiler contains a large quantity of water at high pressure, and that pressure is suddenly relieved, as would happen in case of rupture, a considerable portion of this large volume of water will be turned instantly into steam, and the resulting explosion will ensue.

When a fracture starts in a boiler-plate, the steam escaping through the rent or opening tends to diminish the pressure rapidly within the boiler; and this causes the rapid formation of a large amount of steam. It must be remembered that the water in the boiler at high pressure is held in the form of water only because of the high pressure exerted on it. If this pressure is relieved, large quantities of water will evaporate into steam at once, without the application of further heat. This almost instantaneous formation of a large quantity of steam prevents the boiler pressure from dropping, and the fracture naturally widens. The larger the body of hot water, the greater the disaster. This accounts for the relative safety of water-tube boilers. The division of the water in such a boiler into small masses in different sections, prevents a violent explosion. Should a water tube burn out, probably nothing more serious would happen than the rapid escape of a considerable quantity of steam, which might fill the boiler-room, drive out the attendants, and ultimately cause the destruction of the boiler because of the absence of water together with a hot fire. It would be necessary for several water tubes to burst at once in order that there should be serious damage from such an accident.

Energy. The available energy in one pound of hot water at 150 lbs. absolute pressure and 358° F., is about 42,800 foot-pounds; that is, it is sufficient to move one pound nearly eight miles; and if at 250 lbs. pressure, it has sufficient energy to move it nearly twelve miles. This energy may be determined somewhat as follows: From the table of the properties of saturated steam, given in the back of the book, it is seen that at 150 lbs. absolute pressure (approximately 135 gauge), the temperature is 358.26° F. The heat contained in a pound of hot water at this temperature will be 330 B. T. U., equivalent to 330 × 778 = 256,740 foot-pounds. This represents the total heat energy in one pound of hot water at boiler pressure; but since one pound of steam at atmospheric pressure contains very many more heat units than a pound of water at 150 lbs. pressure, it is apparent that only a portion of this water can evaporate into steam, the balance remaining as hot water. About 17 per cent of the total energy will be thus available in vaporizing the water into steam; or, approximately, 42,800 foot-pounds per pound of water will be developed. The remaining heat is in the form of hot water.

A cylindrical boiler 5 feet in diameter and 16 feet long is likely to contain about 6,600 pounds of water and 22 pounds of steam. Neglecting the energy of the steam, which is relatively small, the energy in the water due to its expansion from water at boiler pressure into steam at atmospheric pressure, will be approximately 6,600 × 42,800 = 282,480,000 foot-pounds, or 141,240 foot-tons.

A marine boiler 13 feet in diameter and 12 feet long would develop approximately twice this energy, which would be about equivalent to the energy developed by the explosion of a ton of gunpowder. The explosion of one boiler on a modern battleship would develop sufficient power to lift the ship completely out of the water. Of course it must be realized that a large part of this energy is lost, and considerable is consumed in the destruction of the boiler itself, which leaves but a comparatively small amount to be expended in wrecking the immediate surroundings; but it nevertheless is a fact that the energy developed in the explosion of a large boiler is almost beyond the power of comprehension.

Causes of Explosions. Boiler explosions are usually the result of low-water, grease, or scale. The two latter, by preventing the transmission of heat from the water, are likely to cause undue overheating of the furnaces or tubes, which may result in their collapse; these two causes—grease and scale—have been discussed under the subject of "Incrustation."

Low water may be caused by failure of the water glass to indicate properly the amount of water in the boiler, or by failure of the feed pump to work properly.

Safety-valves have been known to be rusted to their seats so tightly that they failed to work at the proper time.

It is seldom that a boiler can fail as the result of defective design, for the laws in regard to construction, especially of marine boilers, are very definite. Defective workmanship or material, however, cannot be easily discovered; and it is possible that corrosion or incrustation may take place locally, without being readily detected; and, indeed, boiler-plates may even be tapped, and their thickness calipered, without discovering small local weaknesses which later may cause disaster. Minute fractures which escaped the Inspector's detection have later become serious. The majority of explosions can undoubtedly be traced to mismanagement in either care or operation.

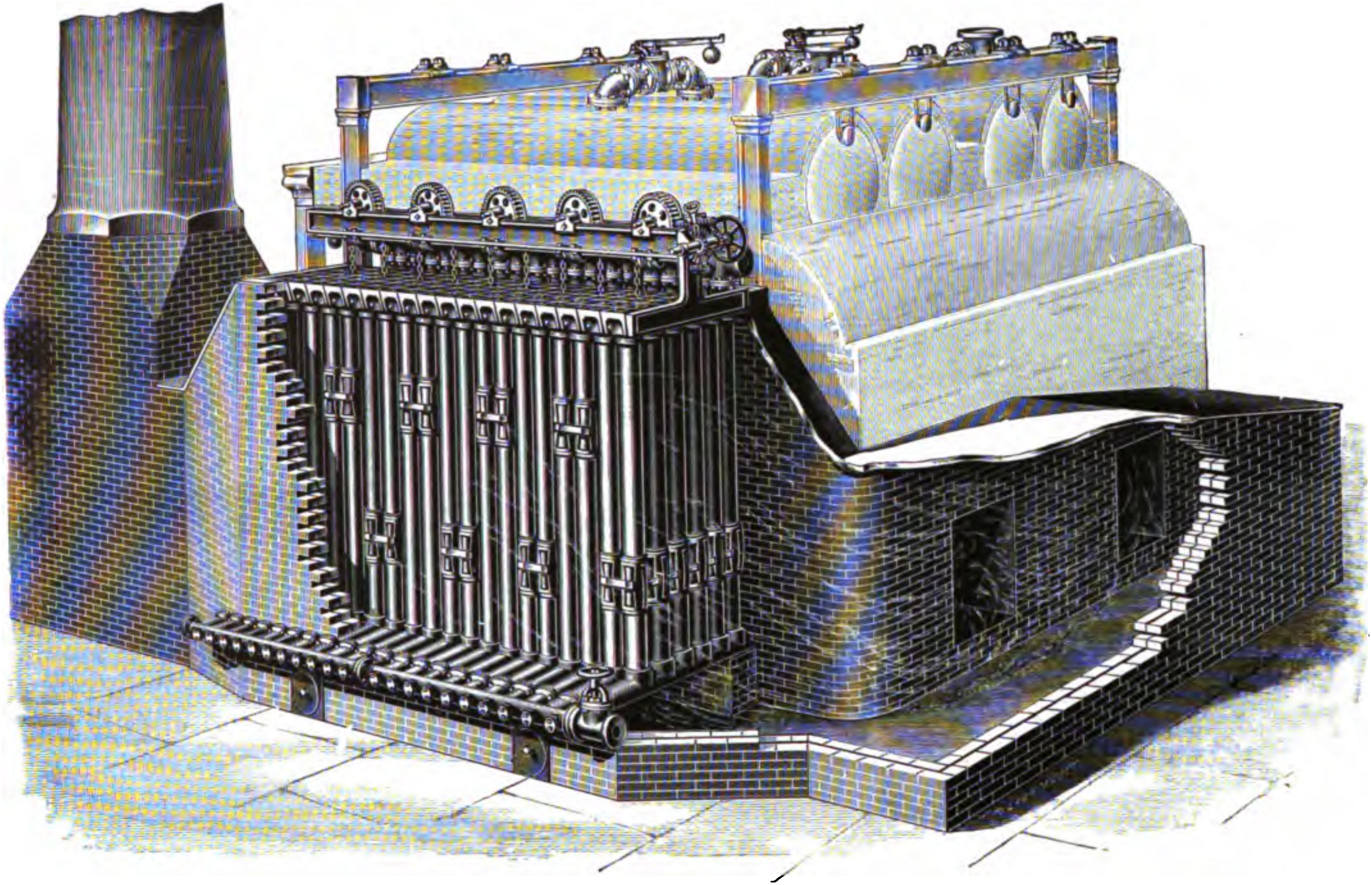
Defective Design. If a boiler is improperly set; if the stays are too small, too few, or cut or bent to clear floats, pipes, etc., danger is likely to result therefrom. All manholes, large handholes, or domes should be strengthened with a reinforcing plate to make up for the material cut out. If the boiler is set too rigidly on its seating, without proper provision for its expansion, trouble is likely to follow. A defective water circulation is likely to cause excessive incrustation and unequal expansion of the plating, which is liable to open seams and produce fractures in the plates.

Deterioration. The strength of a boiler is likely to be impaired by fractures, general corrosion, pitting, or grooving. But external corrosion is the cause of many disasters. It proceeds unnoticed in many cases, and rupture may occur when least expected. In the discussion of "Corrosion," it was shown that improper setting of the boiler would cause or at least aggravate external corrosion; and that, on account of the close setting of the boiler, it was not easy to get at the plates to examine them. The strength of a boiler originally sufficient to sustain high pressure may become suddenly reduced by overheating or over-straining, either of which weakens the plates. Overheating may be caused by poor circulation, lack of water, or the accumulation of sediment or scale. Over-straining is caused by sudden cooling and contraction, or equally by sudden expansion. In starting the fire in a Scotch boiler—or, in fact, in any boiler with a large quantity of water—care must be taken that the fire is started slowly, or the boiler, becoming overheated locally, will develop excessive strains.

Defects of Workmanship. Defective workmanship is not of so frequent occurrence under present conditions as formerly, when many defects used to be produced by careless punching of plates; but for most boilers, and for all marine boilers at present, punching is prohibited; the holes must be drilled, and the plate edges planed and carefully calked. A rigid inspection of material is required, and there seems little danger of unsatisfactory work. Cheap boilers may of course be subject to various defects, but a good boiler should be free from such troubles. Material may be defective and may not be readily detected; but the careful tests now required, especially in marine work, reduce these possibilities to a minimum.

Mismanagement. The pressure in a steam boiler may rise above that at which the safety-valve has been set to operate, because of corrosion or overloading of the valve. Stop-valves are sometimes placed between the boiler and the safety-valve; but this practice should be condemned, as it is possible that the stop-valve may be closed when the fireman thinks the safety-valve is open to the boiler pressure. If the size or lift of the safety-valve is too small, steam may be generated faster than it can escape, in which case the pressure will rise in spite of the safety-valve. It has been claimed that the blowing-off of the safety-valve when the boiler is under excessive pressure may be the cause of starting an explosion; but the reason why this should be so does not seem to be especially clear, and it seems to be improbable if the opening of the safety-valve is sufficient to cause a reduction in pressure. Safety-valves have sometimes been loaded down temporarily to prevent leakage at working pressure; but such a practice is little short of criminal. If a safety-valve leaks, it should be reground, but under no circumstances should the weight on the lever be altered.

It is a common idea that when the furnace plates become very hot, perhaps heated to redness, due to a lack of water, and the feed is turned on, a violent explosion is sure to follow. Experiments show that when a piece of wrought iron is heated to redness and plunged into a weight of water three or four times greater than that of the iron, a comparatively small quantity of steam is disengaged. There is no reason to believe that this quantity would be greater if the iron were in the form of a boiler than in the form of a plate. If a small quantity of water were admitted to hot plates, the danger would



GREEN FUEL ECONOMIZER

be greater; and while a boiler under this condition might explode, the comparatively small quantity of water in it would make the resulting danger much less than if the boiler were under working conditions.

The following experiments illustrate the action of cold water on hot plates. A boiler 25 feet long and 6 feet in diameter was heated red hot and the feed turned on. No explosion occurred; but the sudden contraction of the overheated plates caused the water to pour out in streams at every seam and rivet-hole as far as the fire-mark extended. In another instance, the water was almost entirely drawn off while the fires were burning briskly. When the remaining water had been converted into steam and all the fusible plugs melted out, water at the rate of 28 gallons per minute in a series of fine jets was played on the hot plates. Such treatment may ruin a boiler for further service, though the boiler may not explode.

That a tough paper or cloth is easily torn when once a tear is started, is a well-known fact. Similarly a boiler-plate may be ruptured at slight pressure if a fracture has been started.

The *position of the fracture* or hole has a great influence on the results. In case a large rent occurs at the top of a cylindrical boiler, the steam and hot water may blow out of the hole; and the boiler, if strongly enough seated to stand the reaction, will remain on its seat. The damage to the boiler would be slight. But suppose the same rent were situated on the under side of the boiler near the ground or floor; the effect would be very different, the reaction of the escaping steam would probably blow the whole boiler through the roof.

Investigation. When an explosion occurs, it should be *investigated,* not only to fix the responsibility where it belongs, but also to provide for and take means to prevent future disasters. It has been customary to attribute all explosions to low water, since it is an easy way to throw the responsibility from the makers or owners upon the fireman, who, even if living, cannot defend himself. In the investigation of an explosion, the weights, shapes, positions, and directions of the scattered pieces should be noted, so that their original places may be known. The original size and shape of the boiler and of the fittings should be known as accurately as possible. The primary rent may be discovered from comparison and from deductions of the directions taken by the heavier pieces. Light pieces will generally take the direction of the escaping steam, while the heavy parts take an opposite direction, that of the reaction. A careful examination of the pieces, noting the age of fractures, thickness of plates, amount of corrosion, condition of plates, etc., will generally show the cause. A test of the plates will in many cases show any softening or yielding to the pressure and excessive thinness caused by bulging.

Prevention. The means taken to prevent boiler explosions from most of the above-mentioned causes, have already been given. It is of primary importance that at the start only a well-designed and well-made boiler should be used. The matter of type is not of so much importance; but it is well to use a sectional boiler in large cities or in buildings where many people are employed. There are many methods, some of which have been discussed, that are taken to prevent deterioration by corrosion, fracture, etc. Proper setting is of great importance in this matter. Mishaps from mismanagement may be greatly lessened by the employment of licensed attendants. A boiler should *never* be in the hands of a man who is not thoroughly competent to run it. The most effective method to prevent explosions is the law of the State, compelling regular, thorough inspection and licensed firemen. The inspection by the Boiler Insurance companies is also an efficient method.

During a period of eleven and one-half years, 70,000 boilers were inspected by Boiler Insurance companies. It was estimated that there were 140,000 in use during that time. Of the inspected boilers, there were 23 explosions and 50 collapses, resulting in 27 deaths from explosions, and 28 deaths from collapses. The explosion rate was 1 in 11,000; and the death rate, 1 in 14,600. The uninsured boilers did not make so good a showing, the death rate being 1 in 5,000 boilers, or about 3 times as high as among the insured boilers.

FUEL

There are various kinds of fuel used in steam production, location, cost, and the exigencies of the case being the deciding factors. Usually the kind of fuel is determined upon, and the boiler designed with that end in view. Sometimes, however, the fuel must be adapted to the boiler.

Coal. Coal is not only the most important fuel, but in many localities the only one available. It is of vegetable origin, being the long-decayed product of ancient forests. Frequently it occurs so mixed with earthy matter as to be of little value; but the supply of good coal is still abundant, and likely to be so for some time to come.

The most important elements in coal are hydrogen, producing 62,000 B. T. U. per pound, and carbon, producing 14,500 B. T. U. per pound. Although several coals may have the same total percentage of combustible material and ash, the heat values may not be the same, because heat value depends upon the amounts of hydrogen and carbon they contain. The heat value of fuels is determined by chemical analysis, or by calorimetric test, and varies for coal from different localities. The following table is compiled from several sources:

ANALYSIS AND HEAT VALUE OF VARIOUS COALS

KIND OF COAL	PER CENT OF ASH	THEORETICAL B. T. U. PER POUND	POUNDS OF WATER EVAPORATED PER POUND (THEORETICAL)
Penn. Anthracite	3.49	14,199	14.70
Penn. Anthracite	2.90	14,221	14.72
Penn. Cannel	15.02	13,143	13.60
Penn. Connellsville	6.50	13,368	13.84
Penn. Semi-bituminous	10.70	13,155	13.62
Penn. Brown	9.50	12,324	12.75
Kentucky Caking	2.75	14,391	14.89
Kentucky Cannel	2.00	15,198	16.76
Kentucky Lignite	7.00	9,326	9.65
Indiana Caking	5.66	14,146	14.64
Indiana Cannel	6.00	13,097	13.56
Maryland Cumberland	13.88	12,226	12.65
Arkansas Lignite	5.00	9,215	9.54
Colorado Lignite	9.25	13,562	14.04
Texas Lignite	4.50	12,962	13.41
Washington Lignite	3.40	11,551	11.96

In practice, no fuel gives its theoretical evaporation value. On account of several losses that are inevitably incurred, heat is radiated from, and conducted away by, the boiler setting. The admission of too much air into the furnace, either through the doors or through cracks in the setting, reduces the theoretical evaporation value. Improper firing causes considerable loss; and errors in design, construction, or setting reduce the efficiency.

The different kinds of coal are too numerous to be easily named, but in general they may be classified as *anthracite* or *bituminous*, commonly called *hard* or *soft* respectively, of which there are various subdivisions.

Anthracite. Anthracite coal consists almost entirely of carbon, but has a small amount of hydrocarbon. Good anthracite is lustrous, hard, flinty, but breaks up easily under high temperature. It burns with very little flame and smoke, and gives an intense heat. It does not ignite so readily as the softer varieties of coal; but once started, the fire requires less attention. It is an excellent fuel where the production of smoke is a decided objection.

Semi-Anthracite. This is a coal between pure anthracite and semi-bituminous. It is not so hard as anthracite, and burns more freely. It is not so compact as anthracite, and burns with a short flame, the anthracite having practically no flame.

Semi-Bituminous. This is the next softer grade of coal. It burns more freely than either anthracite or semi-anthracite, contains more volatile hydrocarbon, and is a valuable coal for steaming purposes.

Bituminous. Bituminous coal forms by far the larger portion of steam coal. It contains a large but varying amount of hyrocarbon or bituminous matter. Unless fired with care, it will produce a considerable amount of smoke and clinkers.

Dry Bituminous. This is a black coal with a resinous luster. It burns freely, and kindles with much less difficulty than the anthracites. It is hard, but is easily splintered. When burning, it gives a moderate amount of flame, with but little smoke, and does not cake. It is found chiefly in Maryland and Virginia.

Caking Bituminous. This contains less carbon and more hydrocarbon than the former class. It is not so black; is more resinous; and, under intense heat, readily forms into a solid, pasty mass. Unless frequently broken up, this pasty mass forms a blanket over the grate, and checks the draft. Caking bituminous is a valuable coal for the manufacture of gas. It is mined chiefly in the Mississippi valley.

Cannel. *Cannel* or *long-flame bituminous* coal produces a considerable quantity of smoke. It is mined chiefly in Pennsylvania, Indiana, and Missouri; and is a free-burning coal, with a strong tendency to cake. It is largely used for open-grate purposes.

Lignite. *Lignite*, or *brown coal* is intermediate between coal and peat. It is made up mostly of carbon, with some moisture and mineral matter. Poor varieties are of little value. Good lignite kindles with ease, and burns freely, but is likely to contain a considerable amount of water, and unless kept in a dry place will absorb moisture. It is not a very good fuel, but is used in some localities where other varieties are more expensive. It comes largely from Colorado, Texas, and Washington.

Peat. This is a form of fuel consisting of decayed roots, tree-trunks, etc., and earthy matter. It is found in swamps and bogs, and has been in process of decomposition a much shorter time than any of the coals. It is cut out in blocks and dried. Peat has a specific gravity of .4 to .5, but it can be compressed to a much greater density. It is necessary that peat should be kept in a dry place, for it will readily absorb moisture.

Coke. This is made by driving off by heat the hydrocarbon of bituminous or semi-bituminous coals. It may be made in gas retorts, as a by-product of gas production; or it may be made in coking ovens, the gas being the by-product. The latter form of coke is more valuable as a fuel. If the coal is very moist, or if steam is used in the coking process, as in the manufacture of water gas, the sulphur is burned out. Coke burns without flame; and, with a free supply of air, will make an intensely hot fire.

Charcoal. Charcoal is practically never used for steam fuel, its chief use being for household or manufacturing purposes. It is made by evaporating the volatile matter from wood, either by partial combustion or by heating in retorts. About 50 bushels of charcoal can be obtained from a cord of wood.

Culm. This is a name given to refuse dust at the coal mines, sometimes called *slack*. It can be bought at the mines at a very low rate; but the cost of transportation prohibits its use except in the immediate vicinity of the mines. On account of its fineness, it cannot be burned in an ordinary grate, and is usually blown into the boiler with a sufficient quantity of air, where it burns somewhat like a gas. A grate beneath usually contains a moderate fire, which keeps the culm well ignited and prevents the loss of any particles that might otherwise drop out of the furnace.

Wood. There are two principal divisions of wood—*hardwood*, which is compact and comparatively heavy, such as oak, ash, and hickory; and *soft wood,* which is of soft and porous texture and of less specific gravity, such as pine, birch, and poplar. Wood contains considerable moisture, even if left to season in a dry place; and after being thoroughly dried, it will absorb and retain from 10 to 20 per cent of moisture. Kiln-dried wood contains nearly 8,000 B. T. U. per pound, while the average wood, containing about 25 per cent of moisture, has a heating value of about 6,000 B. T. U.

The chemical composition of different woods is nearly the same, and pound for pound one class of wood contains about the same heating value as another. Pine weighs about half as much as oak per cubic foot, and a cord of such wood contains about half the heating value that a cord of oak would contain.

Sawdust and *shavings* are frequently used as fuel in sawmills and planing mills. This kind of fuel is blown into the furnace with air from a fan, and makes an intense heat. A fine grate at the bottom collects the burning embers, which might otherwise drop into the ash-pan. In mills where sawdust and shavings are used, they are a by-product.

Straw. Threshing machines through the West use straw almost entirely for fuel. It gives an intense heat, furnishing 5,000 to 6,000 heat units per pound; and this is a quick and easy way to get rid of it.

Bagasse is the fibrous portion of the sugar-cane left after the juice has been extracted. In the modern process of sugar manufacture, the cane is pressed so tightly that it is ready for fuel without further treating. Under favorable conditions it forms an excellent fuel. The pressed cane is a by-product which must in some way be got rid of. It is usually fed into the furnace through an automatic hopper; or it may be dumped in the fire-room and fed into the furnace by hand. The furnace is constructed of brick, independent of the boilers; and when bagasse is consumed at a high temperature, the oxygen contained in it is nearly sufficient to satisfy the carbon and hydrogen, so that little air from the outside is required. Such material, of course, cannot be fed into an ordinary furnace.

Liquid Fuels. These consist of petroleum and its products, and their use has become quite extensive in the last few years. The field would undoubtedly be wider were there less difficulty in obtaining a regular and constant supply. The greatest quantities of petroleum oil are produced in the United States and Russia. Large quantities are found on the Pacific Coast, especially in Southern California; and in that section of the country, oil is used as fuel to a greater extent than in the East, being largely used on tugboats, ferryboats, and locomotives.

The following, approximately, is the composition of petroleum:

Carbon	82 to 87 per cent.
Hydrogen	11 to 15 per cent.
Oxygen	$\frac{5}{10}$ to 6 per cent.

The theoretical heating power of petroleum is approximately 20,000 B. T. U. per pound, which is nearly half as much again as that of good coal. Oil has a further advantage over coal, in that no unburned fuel necessarily passes through the furnace, and there is no ash—an important item in marine work.

The composition and specific gravity of petroleums vary considerably, many of the lower grades being unsafe on account of their low flash-point.

The fuel is fed into the furnace through an atomizer operated either by steam or by compressed air. Several types of such devices are shown in Fig. 72. The use of the oil as a fuel can be readily controlled by the simple manipulation of a valve; and if the fire is once regulated to produce the required heat, it can be kept at that point with very little care. The fire can be started with slight trouble, and can be extinguished instantly. The vaporizing efficiency of oil is much greater than that of coal; and on the Pacific Coast, where oil can be readily obtained, it is a much more economical fuel. If burned properly, without too heavy an air-blast, there should be no production of smoke. A considerable saving may be effected in the

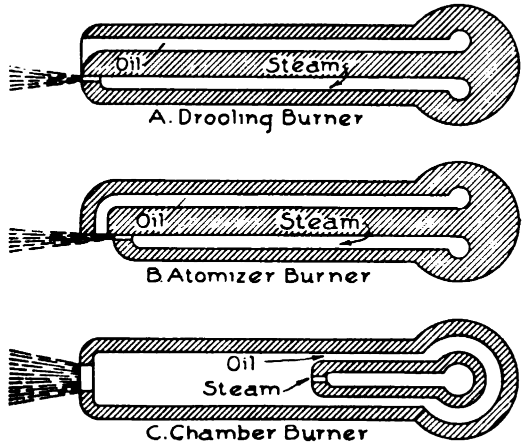


Fig. 72. Types of Atomizers for Liquid Fuel.

fire-room force, one man being able to operate several burners. There is, of course, danger from explosion, on account of vapor which rises from the fuel; but if the fuel tank is thoroughly ventilated, there is little danger from this source.

Oil fuel may be used to advantage in what is called *mixed firing*; that is, the oil may be sprayed onto the bed of burning coal. This has been condemned by many engineers, but it has nevertheless gained considerable headway, and, under proper conditions, has given satisfactory results. It is beyond the scope of this work to go minutely into the subject of oil fuel; but for further information the student is referred to the reports of the Oil Fuel Boards of the U. S. Navy and of the British Admiralty.

Gas. Gas has many advantages over any other kind of fuel. There are four different varieties—*natural gas*, *coal gas*, *water gas*, and *producer gas*. *Natural gas* is used largely in the vicinity of Pittsburg, Buffalo, and some parts of Indiana, both for illuminating and for steam purposes. Where natural gas is plentiful, it is by far the cheapest fuel that can be used.

Coal gas, made by the distillation of coal, and *water gas*, obtained by the decomposition of steam by incandescent carbon, have been used both for lighting and for fuel; but in most cases these gases may be used to greater economy directly in the cylinder of a gas engine than as fuel under a steam boiler. The same may be said of *producer gas*, which is made by blowing steam and air through incandescent coal.

The relative values of these gases for evaporation, are shown in the following table:

EVAPORATIVE POWER OF GASES

	NATURAL GAS	COAL GAS	WATER GAS	PRODUCER GAS
Cubic feet of gas	1,000	1,000	1,000	1,000
Pounds of water evaporated	893	591	262	115

Experiments in Pittsburg have shown that 1,000 cubic feet of natural gas equals 80 to 133 pounds of coal. The coal used in the comparison varied from 12,000 to 13,000 B. T. U. per pound.

The Western Society of Engineers has stated that one pound of good coal is equivalent in heating value to 7½ cu. ft. of natural gas.

As in the case of petroleum, the economy of burning gaseous fuels depends upon the locality.

Artificial Fuels. The waste of charcoal, coal, sawdust, etc., is often pressed into cakes or *briquettes*, by means of some adhesive mixture, with compression. Wood tar, coal tar, and clay are used, according to convenience. These cakes are compact, can be stored in small space, and are used where good fuels are difficult to obtain.

STEAM BOILER TRIALS

The object of a boiler trial is to determine the quantity and quality of steam that the boiler will supply under given conditions, the horse-power of the boiler, the amount of fuel it takes to make the required steam, and its efficiency.

The quantity of steam is taken as the amount of water evaporated, which, of course, is the total amount fed into the boiler during the test the water-level being the same at the beginning and the end.

The quality of the steam can be determined by some form of calorimeter already described; and the efficiency is the ratio of the heat units utilized in evaporating the water to the total heat supplied to the boiler. The heat utilized in evaporation can be found by multiplying the number of pounds of feed-water by the number of heat units required to change the water at the temperature of the feed into steam at gauge pressure. The heat units supplied can be determined by carefully weighing the fuel used during the test, and deducting the amount of ash and unburned fuel going through the grates, with proper allowance for moisture, multiplying the result by the total heat of combustion of the fuel. The heat of combustion can be obtained by calculation, or by means of a fuel calorimeter.

Under a short test the boiler must be in good working order and fired for some hours before the beginning of the test, so that the brickwork and chimney may be thoroughly heated. Shortly before the test is begun, the fire may be allowed to burn low; and by reducing the amount of steam taken from the boiler, the pressure can be kept constant. The fire may then be drawn, the grate cleaned, and a new fire quickly started, with wood and fresh coal. Toward the end of the test the fire may be allowed to burn low, and at the close may be drawn and quenched with water, the unburned fuel being allowed for. In a long test of twenty-four hours or more, this is not necessary.

If the boiler is fed by a steam pump, the pump should be run by steam taken from some other boiler, if convenient; if not, the amount of steam used by the pump must be determined and allowed for. If the feed-water is supplied by an injector, it will take steam from the boiler itself. About 2 per cent of this steam is consumed in forcing the water into the boiler, the remainder going to heat the feed-water.

During the boiler trial, observations of temperatures and pressures should be made at the same time, and at about 15-minute intervals. In order to obtain the result of the test, the following must be known:

1. Amount (in pounds) of coal burned, and number of pounds of ashes left;
2. Number of pounds of water pumped into boiler;
3. Temperature of feed-water when it enters boiler;
4. Pressure of steam in boiler;
5. Quality of steam discharged from boiler—that is, the per cent of moisture in the steam.

The coal for the furnace can be conveniently weighed in barrels, and may be fired directly from these barrels or dumped on the fire-room floor. The barrels should be carefully weighed when full and empty, and the time recorded, so that there may be no possibility of counting one barrel twice or omitting any. The rate of combustion will be fairly uniform, and the calculations at the times of emptying the barrel will fairly indicate whether or not an error has been made. Any unburned coal should be weighed, and the amount subtracted.

The condition of the fire for a twenty-four-hour test should be the same at the beginning and the end. This condition is estimated by the eye; and unless great care is used, an appreciable error is likely to be made If the coal consumption is 15 to 20 lbs. per square foot of grate surface, an error of two inches in estimating the thickness of the fire may cause an error of as much as 2 per cent in the final results.

The wood used in starting the fire should be carefully weighed, and may be considered as equal to $\frac{4}{10}$ of the same weight of coal. The clinker and ashes should be carefully collected and weighed, and a sample of the ashes examined to obtain the amount of unburned fuel.

There are several ways of determining the amount of water pumped into the boiler. The best method is to weigh it in tanks or barrels set upon standard scales. There should be two or more barrels of sufficient size, so that the filling and emptying may not be hurried. They should be set high enough to discharge readily into the tank or hot well from which the feed-water is drawn. The valves should be large and should open quickly, so that the emptying may not be delayed. If barrels are used, they should be numbered, and the weight of each accurately noted, so that there may be no mistake in deducting the weight of a barrel from the total weight of barrel and water. When one barrel is being emptied, the other may be filled. The weigher must use care and intelligence; otherwise he may become confused in his records, as in a boiler of considerable size the barrels fill and empty rapidly. At the beginning of the test, the level of the water in the hot well should be recorded, and at the end of the test should be brought to the same mark. If inconvenient to weigh the water, it may be measured by a meter; but if a meter is used, it should be tested and its error determined under like conditions of temperature and pressure. The feed-water should be free from air, as otherwise too large a meter reading will be recorded.

The level in the water-glass of the boiler should be carefully noted at the beginning and end of the test. If possible, the level should be constant throughout the test; and if there is any difference between the beginning and the end, due allowance should be made for it.

The temperature of the feed-water can be taken best by means of a thermometer in a cup filled with oil screwed into the feed-pipe near the check-valve. If the temperature is nearly constant, readings at 15-minute intervals will suffice; otherwise readings should be taken every five minutes.

The steam pressure shown by the gauge should be as nearly constant as possible throughout the test, and should be practically the same both at the beginning and at the end. Gauge readings should be recorded every 15 minutes, and the fireman should see that the pressure is constant. The gauge should be tested, and corrected if necessary.

Barometric readings should also be taken, two or three being sufficient for a ten-hour run. These readings, in inches, may be made to indicate pounds pressure by multiplying by .491, this being the weight of one cubic inch of mercury. If the trial is on a vertical boiler which furnishes superheated steam because of the heat being in contact with the tubes above the water-level, both the pressure-gauge and the thermometer should be used, so that the amount of superheating can readily be found by subtracting the temperature due to pressure (obtained from the steam tables) from the temperature readings.

The quality of steam can readily be determined by a calorimeter. If there is sufficient steam space within the boiler, from 1 to 2 per cent priming will generally result. If the steam space is inadequate, there will be more priming. If more than 2 per cent priming is present, the steam will blow white from the gauge-cocks when opened; if less than 2 per cent, it will appear blue.

The above observations are of the more important class, and *must* be taken. In addition to these, it is well to take samples of the flue gas at intervals and from various places in the furnace or chimney, the object being to determine whether there is a sufficient supply of air admitted, or whether there is too much. The draft of the chimney may be measured by means of a U-tube partially filled with water, or by a draft-gauge.

It is well to bear in mind that in making the boiler test the utmost care must be used, both in taking observations and in recording them, and in working up the results of the trial. A committee of the American Society of Mechanical Engineers has recommended a code of rules for boiler trials, and the following standard form for recording results. These are too voluminous for complete reproduction, and they can be found in full in Vol. XXI of the *Proceedings* of the above Society for the year 1900. The following code of rules is practically an abstract of the above-mentioned code:

PRELIMINARIES TO A TEST

1. In preparing for and conducting trials of steam boilers, the specific object of the proposed trial should be clearly defined and steadily kept in view.

2. Measure and record the dimensions, position, etc., of grate and heating surfaces, flues, and chimneys; proportion of air-space in the grate-surface; kind of draught, natural or forced.

3. Put the boiler in good condition. Have heating surface clean inside and out; grate-bars and sides of furnace free from clinkers; dust and ashes removed from back connections; leaks in masonry stopped; and all obstructions to draught removed. See that the damper will open to full extent, and that it may be closed when desired.

Test for leaks in masonry by firing a little smoky fuel and immediately closing damper. The smoke will escape through the leaks if there be such.

4. Have an understanding with the parties in whose interest the test is to be made, as to the character of the coal to be used. The coal must be dry; or, if wet, a sample must be dried carefully, and a determination of the amount of moisture in the coal must be made, the calculation of the results of the test being corrected accordingly. Wherever possible, the test should be made with standard coal of a known quality. For that portion of the country east of the Alleghany mountains, good anthracite egg coal or Cumberland semi-bituminous coal may be taken as the standard for making tests. West of the Alleghany mountains and east of the Missouri river, Pittsburg lump coal may be used.

In all important tests, a sample of coal should be selected for chemical analysis.

5. Establish the correctness of all apparatus used in the test for weighing and measuring. These are: *1.* Scales for weighing coal, ashes, and water. *2.* Tanks or water-meters for measuring water. Water-meters, as a rule, should only be used as a check on other measurements. For accurate work the water should be weighed or measured in a tank. *3.* Thermometers and pyrometers for taking temperatures of air, steam, feed-water, waste gases, etc. *4.* Pressure-gauges, draft-gauges, etc.

6. Before beginning a test, the boiler and chimney should be thoroughly heated to their usual working temperature. If the boiler is new, it should be in continuous use at least a week before testing, so as to dry the mortar thoroughly and heat the walls.

7. Before beginning a test, the boiler and connections should be free from leaks; and all water connections, including blow and extra feed-pipes, should be disconnected or stopped with blank flanges, except the particular pipe through which water is to be fed to the boiler during the trial. In locations where the reliability of the power is so important that an extra feed-pipe must be kept in position, and in general when, for any other reason, water-pipes other than the feed-pipes cannot be disconnected, such pipes may be drilled so as to leave openings in their lower sides, which should be kept open throughout the test as a means of detecting leaks or accidental or unauthorized

opening of valves. During the test the blow-off pipe should remain exposed.

If an injector is used it must receive steam directly from the boiler being tested, and not from a steam-pipe or from any other boiler.

See that the steam pipe is so arranged that water of condensation cannot run back into the boiler. If the steam pipe has such an inclination that the water of condensation from any portion of the steam-pipe system may run back into the boiler, it must be trapped so as to prevent this water getting into the boiler without being measured.

8. A test should last at least ten hours of continuous running, and twenty-four hours whenever practicable.

9. The conditions of the boiler and furnace in all respects should be, as nearly as possible, the same at the end as at the beginning of the test. The steam pressure should be the same, the water-level the same, the fire upon the grates should be the same in quantity and condition, and the walls, flues, etc., should be of the same temperature. To secure as near an approximation to exact uniformity as possible in conditions of the fire and in temperatures of the walls and flues, the following method of starting and stopping a test should be adopted.

10. **Standard Method.** Steam being raised to the working pressure, remove rapidly all the fire from the grate, close the damper, clean the ash-pit, and as quickly as possible start a new fire with weighed wood and coal, noting the time of starting the test and the height of the water-level while the water is in a quiescent state, just before lighting the fire.

At the end of the test, remove the whole fire, clean the grates and ash-pit, and note the water-level when the water is in a quiescent state; record the time of hauling the fire as the end of the test. The water-level should be as nearly as possible the same as at the beginning of the test. If it is not the same, a correction should be made by computation, and not by operating pump after test is completed. It wil generally be necessary for a time to regulate the discharge of steam from the boiler tested, by means of the stop-valve, while fires are being hauled at the beginning and at the end of the test, in order to keep the steam pressure in the boiler at those times up to the average during the test.

11. **Alternate Method.** Instead of the Standard method above

described, the following may be employed where local conditions render it necessary:

At the regular time for slicing and cleaning fires, have them burned rather low, as is usual before cleaning, and then thoroughly cleaned; note the amount of coal left on the grate as nearly as it can be estimated; note the pressure of steam and the height of the water-level—which should be at the medium height to be carried throughout the test—at the same time; and note this time as the time of starting the test. Fresh coal, which has been weighed, should now be fired. The ash-pits should be thoroughly cleaned at once after starting. Before the end of the test the fires should be burned low, just as before the start, and the fires cleaned in such a manner as to leave the same amount of fire, and in the same condition, on the grates as at the start. The water-level and steam pressure should be brought to the same point as at the start, and the time of the ending of the test should be noted just before fresh coal is fired.

12. **Keep the Conditions Uniform.** The boiler should be run continuously, without stopping for meal-times or for rise or fall of pressure of steam due to change of demand for steam. The draught, being adjusted to the rate of evaporation or combustion desired before the test is begun, should be retained constant during the test, by means of the damper.

If the boiler is not connected to the same steam pipe with other boilers, an extra outlet for steam with valve in same should be provided, so that in case the pressure should rise to that at which the safety-valve is set, it may be reduced to the desired point by opening the extra outlet, without checking the fires.

If the boiler is connected to a main steam pipe with other boilers, the safety-valve on the boiler being tested should be set a few pounds higher than those of the other boilers, so that in case of a rise in pressure the other boilers may blow off, and the pressure be reduced by closing their dampers, allowing the damper of the boiler being tested to remain open, and firing as usual.

All the conditions should be kept as nearly uniform as possible, such as force of draught, pressure of steam, and height of water. The time of cleaning the fires will depend upon the character of the fuel, the rapidity of combustion, and the kind of grates. When very good coal is used, and the combustion not too rapid, a ten-hour test may be run without any cleaning of the grates other than just before the beginning and just before the end of the test. But in case the grates have to be cleaned during the test, the intervals between one cleaning and another should be uniform.

13. **Keeping the Records.** The coal should be weighed and delivered to the fireman in equal portions, each sufficient for about one hour's run; and a fresh portion should not be delivered until the previous one has all been fired. The time required to consume each portion should be noted, the time being recorded at the instant of firing the first of each new portion. It is desirable that at the same time the amount of water fed into the boiler be accurately noted and recorded, including the height of the water in the boiler and the average pressure of steam and temperature of feed during the time. By thus recording the amount of water evaporated by successive portions of coal, the record of the test may be divided into several divisions, if desired, at the end of the test, to discover the degree of uniformity of combustion, evaporation, and economy at different stages of the test.

14. **Priming Tests.** In all tests in which accuracy of results is important, calorimeter tests should be made of the percentage of moisture in the steam, or of the degree of superheating. At least ten such tests should be made during the trial of the boiler, or so many as to reduce the probable average error to less than one per cent; and the final records of the boiler test should be corrected according to the average results of the calorimeter tests.

On account of the difficulty of securing accuracy in these tests, the greatest care should be taken in the measurements of weights and temperatures. The thermometers should be accurate within a tenth of a degree; and the scales on which the water is weighed, to within one-hundredth of a pound.

15. As each fresh portion of coal is taken from the coal-pocket, a representative shovelful should be selected from it and placed in a barrel or box, to be kept until the end of the trial, for analysis. The samples should then be thoroughly mixed and broken. This sample should be put in a pile, and carefully quartered. One quarter may then be put in another pile, and the process repeated until five or six pounds remain. One portion of this sample is to be used for the

determination of the moisture and heating value; the other, for chemical analysis.

16. The ashes refuse should be weighed dry, and a sample frequently taken to show the amount of combustible material passing through the grate. To get a representative ash sample, the ash-pile should be quartered as required for the coal.

17. The quality of the fuel should be determined by heat test, by analysis, or by both.

18. The analysis of the flue gases is an especially valuable method of determining the relative value of different methods of firing or of different kinds of furnaces. Great care should be taken to procure average samples, since the combustion of the gases may vary at different points in the flue; and as the combustion of flue gas is liable to vary from minute to minute, the sample of gas should be drawn through a considerable period of time.

19. It is desirable to have a uniform system of determining and recording the quantity of smoke produced. This is usually expressed in percentages, depending upon the judgment of the observer.

20. In tests for the purpose of scientific research in which the determination of all variables is desirable, certain observations should be made which in general are not necessary—such as the measurement of air-supply, the determination of its moisture, the determination of the heat loss by radiation, the infiltration of air through the setting, etc.—but as these determinations are rarely undertaken, no definite instructions are here given.

21. Two methods of defining and calculating the efficiency of the boiler are recommended. They are:

$$(1)\ \text{Efficiency of the boiler} = \frac{\text{Heat absorbed per pound of combustible}}{\text{Calorific value of one pound of combustible}}$$

$$(2)\ \text{Efficiency of boiler and grate} = \frac{\text{Heat absorbed per pound of coal}}{\text{Calorific value of one pound of coal}}$$

The first of these is the one usually adopted.

22. An approximate statement of the distribution of the heating value of the coal among the several items of heat utilized, may be included in the report of a test when analyses of the fuel and chimney gases have been made.

23. **Record of the Test.** The data and results of the trial should be recorded in a systematic manner, according either to Table 1

(see Vol. XXI, Transactions of the American Society of Mechanical Engineers), or Table 2, taken from those "Transactions."

TABLE 2

Data and Results of Evaporative Test

Arranged in accordance with the short form advised by the Boiler Test Committee of the American Society of Mechanical Engineers, Code of 1899:

Made by .... on .... boiler, at ....
To determine ....
Kind of fuel ....
Kind of furnace ....
Method of starting and stopping the test (*Standard* or *Alternate*, Arts. X and XI, Code)
Grate surface .... sq. ft.
Water-heating surface .... " "
Superheating surface .... " "

Total Quantities

1. Date of Trial ....
2. Duration of Trial .... hours
3. Weight of coal as fired .... lbs.
4. Percentage of moisture in coal .... per cent
5. Total weight of dry coal consumed .... lbs.
6. Total ash and refuse .... "
7. Percentage of ash and refuse in dry coal .... per cent
8. Total weight of water fed to boiler .... lbs.
9. Water actually evaporated, corrected for moisture or superheat in steam .... "
10. Equivalent water evaporated into dry steam from and at 212° F. .... "

Hourly Quantities

11. Dry coal consumed per hour .... lbs.
12. Dry coal per square foot of grate surface per hour .... "
13. Water evaporated per hour corrected for quality of steam .... "
14. Equivalent evaporation per hour from and at 212° F. .... "
15. Equivalent evaporation per hour from and at 212° F. per square foot of water-heating surface .... "

Average Pressures, Temperatures, Etc.

16. Steam pressure by gauge .... lbs. per sq. in.
17. Temperature of feed-water entering boiler ...... degrees
18. Temperature of escaping gases from boiler ..... "
19. Force of draught between damper and boiler ... ins. of water
20. Percentage of moisture in steam, or number or degrees of superheating .... per cent or degrees

Horse-Power

21. Horse-power developed (item 14 ÷ 34½) . . . . H. P.
22. Builder's rated horse-power . . . . "
23. Percentage of builder's rated horse-power developed . . . per cent.

Economic Results

24. Water apparently evaporated under actual conditions per pound of coal as fired (item 8 ÷ item 3) . . . . lbs.
25. Equivalent evaporation from and at 212° F. per pound of coal as fired (item 10 ÷ item 3) . . . . "
26. Equivalent evaporation from and at 212° F. per pound of dry coal (item 10 ÷ item 5) . . . . "
27. Equivalent evaporation from and at 212° F. per pound of combustible [item 10 ÷ (item 5 – item 6)] . . . . "

If items 25, 26, and 27 are not corrected for quality of steam, the fact should be stated.

Efficiency

28. Calorific value of the dry coal per pound . . . . B. T. U.
29. Calorific value of the combustible per pound . . . . "
30. Efficiency of boiler (based on combustible) . . . . per cent.
31. Efficiency of boiler, including grate (based on dry coal) "

Cost of Evaporation

32. Cost of coal per ton of —— lbs. delivered in boiler-room $
33. Cost of coal required for evaporating 1,000 lbs. of water from and at 212° F. . . . .

A *log* of the test should be kept on properly prepared blanks containing headings as follows:

TABLE NO. —

Time	Pressures			Temperatures					Fuel		Feed-Water	
	Barometer	Steam gauge	Draft gauge	External air	Boiler-room	Flue	Feed-water	Steam	Time	Pounds	Time	Lbs. or cu. ft.

FIRING

Starting the Fire. The fireman should first ascertain the water-level; as the gauge-glass is not always reliable, on account of impurities, foam, etc., the gauge-cocks should be tried. In a battery of boilers, the gauge-cocks of each should be opened, for the water may not stand at the same level in each. The safety-valve should be raised slightly from its seat. If the fire has been banked over night, open the draughts, and rattle down the ashes and clinkers from the grate. In case the fire has been allowed to go out, a new one may be started if the gauge-glass shows the proper amount of water, and the valves work well.

If anthracite coal is used, first throw a thin layer of coal all over the grate, then place a piece of wood across the mouth of the furnace just inside the door and lay other pieces of wood at right angles to the cross-piece with the ends resting on it. This allows a space under the wood for air. Now throw on coal until the wood is covered. The fire may be started with oily cotton waste, shavings, or any combustible material.

Keep the furnace door open and the draught-plate closed until the wood is burning freely, which causes the flame to pass over and through the coal and to ignite it. The fire is then spread or pushed back evenly over the furnace bars; the furnace door closed; the ash-pit door opened, as the draught requires; and more coal added when necessary. If bituminous coal is used, do not spread a thin layer over the grate bars under the wood.

The fire at the start should be slow, to cause gradual, uniform heating of the water and various parts of the boiler. If steam is raised too rapidly, enormous strains are set up, due to unequal expansion, thereby causing leakage at joints, and perhaps rupture.

If the boiler is of the water-tube type, steam may be raised more rapidly, because the amount of water is less and the joints are usually placed at some distance from the intense heat of the fire.

The fire being started, the method of adding coal depends upon the fireman, the kind of coal, the type of boiler, and the rate of combustion. There are three general methods of firing—*spreading*, *alternate* or *side firing*, and *coking*.

Spreading is accomplished by placing small amounts of coal uniformly over the entire surface of the grate at short intervals. By this method, the coal is thrown just where it is wanted and then not disturbed. The fire should be hollowed in the center; that is, it should be thicker at the sides. Good results are obtained from this method, since the fire can be kept in the right condition at all times, if the coal is of the right sort. During the operation of firing, the door should be kept open as little as possible, or the fire will be cooled by the entrance of cold air. For a short time, while the coal is giving off gas, the draught-plate of the furnace door should be opened, in order that sufficient air may be admitted above the coal to burn the hydrocarbons.

When the *alternate* or *side firing* method is used, coal is spread so as to cover one side of the fire completely at one firing, leaving the other side bright. At the next firing, the bright side is covered. The hydrocarbons given off by the fresh coal are burned by the hot gases from the incandescent coal. This method is superior to spreading, because the entire furnace is not cooled off by the addition of fresh fuel.

Side firing is most advantageous with two furnaces leading to a common combustion chamber. The furnaces are fired at regular intervals with moderate charges of coal, and the draught-plates are opened while the coal is giving off gas.

The two systems described above are best adapted to anthracite coal, since it burns with comparatively little smoke.

With bituminous coal, which is soft and burns with considerable smoke, the *coking method* is used. The coal is piled on the grate just inside the door, and allowed to coke from 15 to 30 minutes. During this time, the hydrocarbons are driven off and burned by the heat from the fire. In order fully to accomplish this, air must be admitted above the grate through the draught-plates of the furnace door. The coke is then pushed backward over the fire, and a new supply placed on the front of the grate. The air admitted prevents the forming of carbon monoxide gas and smoke. At the same time, however, it cools the furnace somewhat and reduces the rate of evaporation; but this objection is not serious unless a boiler must be worked to its maximum capacity in order to furnish the required amount of steam. If this is the case, economy is sacrificed to rapidity, for a low rate of combustion is usually more economical than a high rate.

The necessary thickness of a bed for the best results, is found by experiment. It depends on the draught and the kind of coal used. If the former is strong, and the coal in large lumps, the bed may be thick (about one foot); but if the draught is weak, or if the coal is small, the bed must be thin (about three or four inches), so that sufficient air may pass through. In marine and locomotive work, with forced draught, the bed must be very thick to get a large coal consumption per square foot per hour. With the same draught, bituminous coal can be fired more thickly than anthracite.

After finding from experiment the best thickness for the bed, keep it at that thickness. Always keep the bed of uniform thickness, and never let the fire burn holes in the bed, and do not let the rear of the grate become bare. If a larger amount of steam is required, fire smaller quantities at more frequent intervals. Do not fire a large amount of coal, and wait for the pressure to rise The firing of fresh coal chills the furnace and temporarily retards combustion. The coal should be fired in small quantities and as quickly as possible. Keep the fire free from ashes and clinkers, but do not clean the fires oftener than is necessary.

Four tools are used for cleaning the fire—the *slice-bar*; the *prick-bar*; the *clinker hook*, sometimes called the *devil's claw*; and the *hoe* or *rake*.

The *slice-bar* is a long, straight bar, with the end flattened. It is used to break up clinkers by thrusting it between the grate and the fire. It is also used to break up caking coal. The *prick-bar* is similar to the slice-bar, except that the end is bent at right angles like a hook. To remove ashes, the prick-bar is run along, up between the grate bars, from underneath. This bar is often made with detachable hook, so that the end may be replaced when burned off. The *clinker hook*, or *devil's claw*, is used to haul the fire forward. The *hoe*, or *rake*, is used to draw out cinders, to haul the fire forward, etc.

In cleaning the fire, the fireman first looks to the water and steam. There should be enough water and sufficient steam pressure to last during cleaning. Then he breaks up the clinkers with the slice-bar, and removes the ashes with the prick-bar. If necessary, he pushes the fire to the rear, thoroughly cleans the front of the grate bars, and then hauls it forward and cleans the back of the furnace bars. Some firemen clean one side at a time, instead of first the front and then the rear. The fire should be allowed to burn down before cleaning; but sufficient fuel, called *chaff*, should be left to start the fire quickly. Before cleaning, partly close the dampers, so that the amount of cold air admitted will be small. For this reason and to prevent loss of pressure, clean as rapidly as possible.

Banking the fire depends upon the condition of the fire, the fireman himself, and the length of time it is to remain banked. First clean and place all the coal in a small space at the bridge; then cover with fresh coal to a depth depending on the length of time the fire is to remain banked. Then close all dampers and open the door. Some firemen cover the front of the furnace bars with ashes.

To start from a banked fire, first examine the condition of the water-level, steam pressure, safety-valves, etc. Then clean the fire with the slice-bar, and rattle down the ashes with the prick-bar. After spreading the coal evenly over the grate, cover with a thin layer of coal, and open the dampers.

CARE OF BOILERS

Any amount of time spent in the proper care of a steam boiler will be amply repaid, for this is of great importance. The boiler, of course, should be so designed and constructed that all parts can be inspected readily; but this is of little benefit unless proper and rigid inspections are made. All internal fittings, such as fusible plugs, water alarms, feed-pipes, and the like, should occasionally be examined to see if they are tight and in good working order. If due care is not given to the boiler, its life will be materially shortened.

The following rules for the management and care of boilers have been established by the Hartford Steam Boiler Inspection & Insurance Company, and should be carefully followed, whether the boiler is insured by the above-mentioned company or not:

1. **Condition of Water.** The first duty of an engineer, when he enters his boiler-room in the morning, is to ascertain how many gauges of water there are in his boilers. Never unbank or replenish the fires until this is done. Accidents have occurred, and many boilers have been entirely ruined from neglect of this precaution.

2. **Low Water.** In case of low water, cover the fires immediately with ashes; or, if no ashes are at hand, use fresh coal, and close ash-pit doors. Do not turn on the feed under any circumstances, nor tamper with or open the safety-valve. Let the steam outlets remain as they are.

3. **In Case of Foaming.** Close throttle, and keep closed long enough to show true level of water. If that level is sufficiently high, feeding and blowing will usually suffice to correct the evil. In case

of violent foaming, caused by dirty water or by change from salt to fresh water or *vice versa*, in addition to the action above stated, check draught, and cover fires with fresh coal.

4. **Leaks.** When leaks are discovered, they should be repaired as soon as possible.

5. **Blowing Off.** Clean furnace and bridge wall of all coal and ashes. Allow brickwork to cool down for two hours at least before opening blow-off. A pressure exceeding 20 lbs. should not be allowed when boilers are blown out. Blow out at least once in two weeks. In case the feed becomes muddy, blow out six or eight inches every day. When surface blow-cocks are used, they should be frequently opened for a few minutes at a time.

6. **Filling Up the Boiler.** After blowing down, allow the boiler to become cool before filling again. Cold water pumped into hot boilers is very injurious, from the sudden contraction set up.

7. **Exterior of Boiler.** Care should be taken that no water comes in contact with the exterior of the boiler, either from leaky joints or from other causes.

8. **Removing Deposit and Sediment.** In tubular boilers, the handholes should be frequently opened, all collections removed, and fore-plates carefully cleaned. Also, when boilers are fed in front and blown off through the same pipe, the collection of mud or sediment in the rear end should be removed frequently.

9. **Safety-Valves.** Raise the safety-valves cautiously and frequently, as they are liable to become fast in their seats and useless for the purpose intended.

10. **Safety-Valve and Pressure-Gauge.** Should the gauge at any time indicate the limit of pressure allowed by the insurance company, see that the safety-valves are blowing off. In case of difference, notify the company's inspector.

11. **Gauge-Cocks, Glass Gauge.** Keep gauge-cocks clear and in constant use. Glass gauges should not be relied on altogether.

12. **Blisters.** When a blister appears, there must be no delay in having it carefully examined and trimmed or patched, as the case may require.

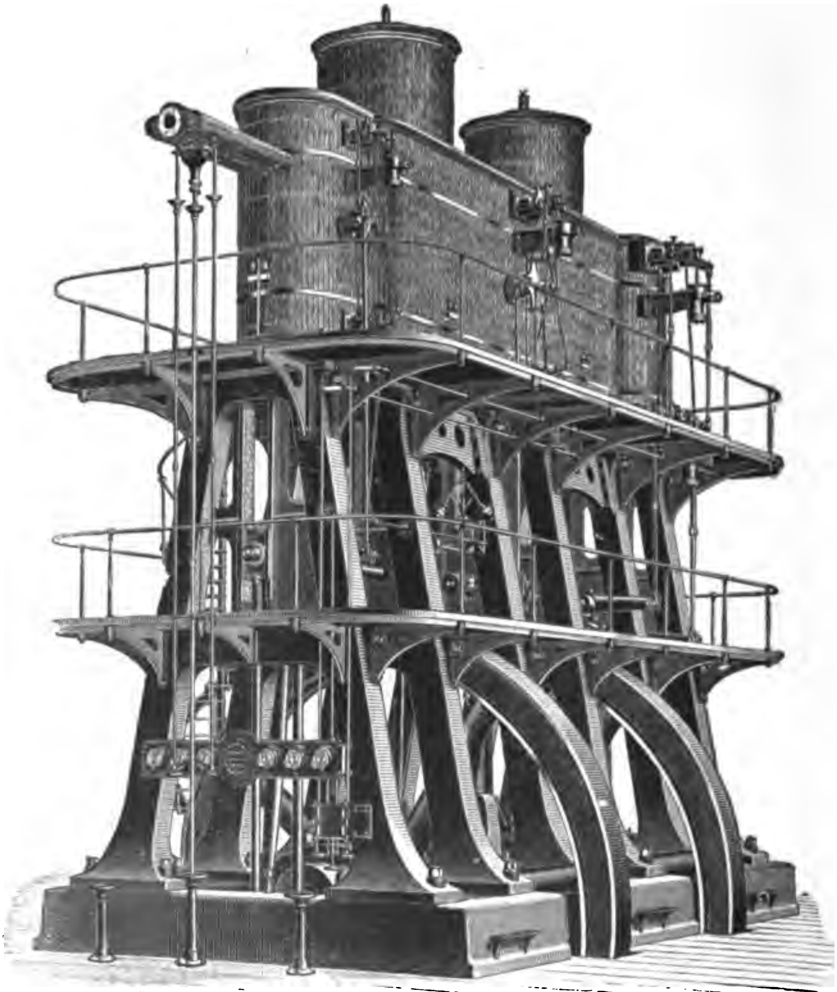
13. **Clean Sheets.** Particular care should be taken to keep sheets and parts of boilers exposed to the fire, perfectly clean; also

all tubes, flues, and connections well swept. This is particularly necessary where wood or soft coal is used for fuel.

14. **General Care of Boilers and Connections.** Under all circumstances, keep the gauges, cocks, etc., clean and in good order, and things generally in and about the engine-room in a neat condition.

15. **Getting Up Steam.** In preparing to get up steam after boilers have been open or out of service, great care should be exercised in making the manhole and handhole joints. Safety-valve should then be opened and blocked open, and the necessary supply of water run in or pumped into the boilers, until it shows at second gauge in tubular and locomotive boilers; a higher level is advisable in vertical tubulars as a protection to the top ends of tubes. After this is done, fuel may be placed upon the grate, dampers opened, and fires started. If chimney or stack is cold and does not draw properly, burn some oily waste or light kindling at the base. Start fires in ample time, so that it will not be necessary to urge them unduly. When steam issues from the safety-valve, lower it carefully to its seat and note pressure and behavior of steam-gauge.

If there are other boilers in operation, and stop-valves are to be opened to place boilers in connection with others on a steam-pipe line, watch those recently fired up, until pressure is up to that of the other boilers to which they are connected; and, when that pressure is attained, open the stop-valves very slowly and carefully.



VERTICAL TRIPLE EXPANSION PUMPING ENGINE
ALLIS-CHALMERS COMPANY

STEAM PUMPS.

Principles of Action. A pump is primarily a machine designed for lifting liquids, or for conveying them to a distance through pipes, or both. To accomplish these results may be used the principles of suction, lifting, or forcing or any combination of them; in practice generally the first is combined with one of the other two.

Suction, so-called, is really the pressure due to the weight of the atmosphere acting to force the liquid into a space wherein a partial vacuum has been created by removing a part of the matter that filled it. The amount of pressure available in any instance for forcing in the liquid, depends upon the completeness with which the matter has been removed from the space in which suction is acting, and on the weight of the atmosphere at that place. If all matter could be removed from the suction chamber, a perfect vacuum would be created and the full pressure of the atmosphere would be effective; a condition closely approaching this is created in the barrel of a pump when no air is present and the plunger or part of the liquid is withdrawn.

If there be any air present, it will expand and fill the vacant space, its pressure falling as the volume increases; the pressure available to force in the liquid by suction will be the difference between that due to the weight of the atmosphere and the final pressure in the suction chamber. The greater the amount of matter removed from the suction chamber, the less will be the resulting pressure therein and the greater the difference available for moving the liquid.

The other factor, the weight of the atmosphere, varies with its condition and the altitude as shown by the barometer. The following table compiled from "Kent's Mechanical Engineers' Pocket Book" and "Nystrom's Mechanics" gives barometer readings, the pressure per square inch, and heights above sea level corresponding to each other; the altitude being at 40° latitude and temperature 60° Fahrenheit.

TABLE I.

Barometric Reading in Inches of Mercury.	Atmospheric Pressure in Pounds per Square Inch.	Altitude above Sea Level in Feet.
30	14.72	0
29.75	14.60	232
29.50	14.47	466
29.25	14.35	703
29.00	14.23	941
28.75	14.11	1181
28.50	13.98	1424
28.25	13.86	1668
28.00	13.74	1915
27.50	13.50	2415
27.00	13.26	2924
26.50	13.02	3443
26.00	12.77	3972
25.50	12.53	4511
25.00	12.27	5061
24.50	12.03	5621
24.00	11.78	6194
23.50	11.54	6778
23.00	11.30	7375
22.50	11.05	7985
22.00	10.80	8609
21.50	10.56	9247
21.00	10.31	9900
20.00	9.81	11254
19.00	9.32	12678
18.00	8.82	14179
17.00	8.33	15766
16.00	7.84	17448
15.00	7.35	19240
14.00	6.86	21155
13.00	6.37	23212
12.00	5.88	25433
11.00	5.39	27848

As the weight of a column of water one foot high and one inch square is 0.433 pound, it follows that the atmospheric pressure, which at sea level is ordinarily 14.72 pounds per square inch, can support a column of water as many feet high as 0.433 is contained in 14.72 or 34.2 feet. The height of column which can be supported decreases as the altitude increases. At the top of a mountain two miles high it would be only 9.81 ÷ 0.433 = 22.7 feet.

This is the height which could be barely supported if the vacuum were perfect. In practice the height to which water can be lifted is much less, because it is impossible to obtain a perfect vacuum on account of leakage of valves and joints, and the atmospheric pressure must overcome friction in pipes and passages, so that the possible lift is reduced to not over 25 feet at sea level and this only for slow working. It is better to attempt only 20 feet or even less if the pump is to work rapidly. If, however, a high suction lift is imperative, it may be obtained by admitting air with the water in suitable proportions to form a heavy spray so that the weight per cubic foot of the mixture is considerably reduced. In this way a lift of as much as 100 feet may be possible. but of course, at the expense of the weight of water pumped per stroke, so that either the pump must be run at a higher speed or a larger pump must be used. The air should be admitted to the suction pipe through several small openings a little above the level of the water in the supply reservoir. The suction must be primed each time the pump is started.

Probably the earliest example of a practical steam pump was that built at Raglan Castle by the Marquis of Worcester in 1630. This employed both suction and pressure, two vessels making the flow continuous. Steam was admitted from a boiler through valve B′, Fig. 1, to the chamber A′, and the pressure forced the water upwards through the valve F′ and pipe E; the valve C′ preventing its return to the reservoir. Meantime the chamber A was shut off from the boiler, and the cold water, condensing the steam, caused a partial vacuum so that water was forced up through valve C and pipe D into the chamber A. When A′ was emptied and A filled, the steam valves were reversed and A′ became filled while A was being emptied.

Practically all pumps, except those to which the liquid comes under pressure, use the suction principle for filling their cylinders. The suction lift is, however, usually but a few feet except where deep mines are drained, and in the case of air pumps for condensers in which the suction is always kept at the highest possible value. In order to improve the suction action, valves and supply pipes must be kept with all seatings and joints absolutely tight.

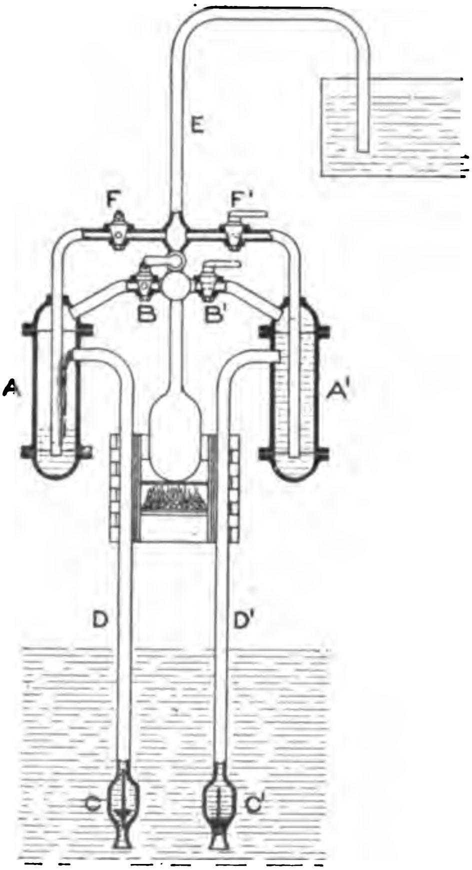


Fig. 1.

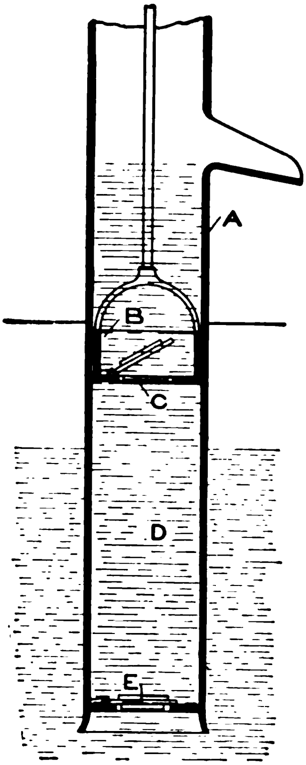


Fig. 2.

Lifting. The lifting pump is simply a bucket B, Fig. 2, working in a vertical barrel A and so arranged that the water is caught above the bucket and raised on the upward stroke to the top of the barrel or to a point where an overflow is provided. Usually the bucket is filled on the down stroke by means of a valve

C in the bottom, and if the lift is for any considerable distance, it is necessary to put a foot valve E at the bottom of the inlet pipe to prevent the liquid from sinking in the barrel when the bucket descends. By this means suction also is introduced, since the space D above the foot valve is filled from the reservoir by suction during the up stroke of the bucket. The pump may then be placed at any distance above the reservoir within the limit already stated, and at any distance horizontally; but it should be remembered that the speed of working and height of lift must be decreased if the horizontal distance is materially increased, and that it is always better to make the suction pipe as short as possible.

Forcing. The suction effect is used in all pumps, at least to fill the barrel, but the lifting principle is uncommon except where the discharge is to be directed from a spout at the top or in the side of the barrel, as in mine drainage and house pumps. If it is necessary to raise the liquid to a point above the pump level, either a piston or a plunger working in a closed chamber is used to force the liquid through discharge valves, against whatever pressure may be necessary, into the delivery pipe; return currents being prevented by the closing of the discharge valves.

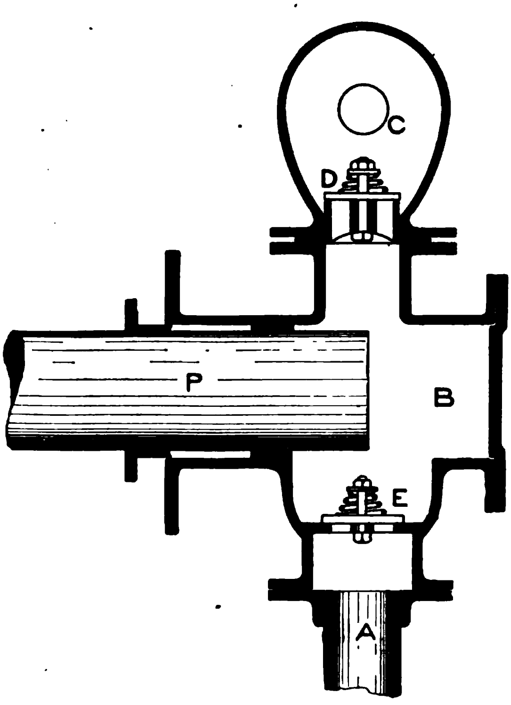


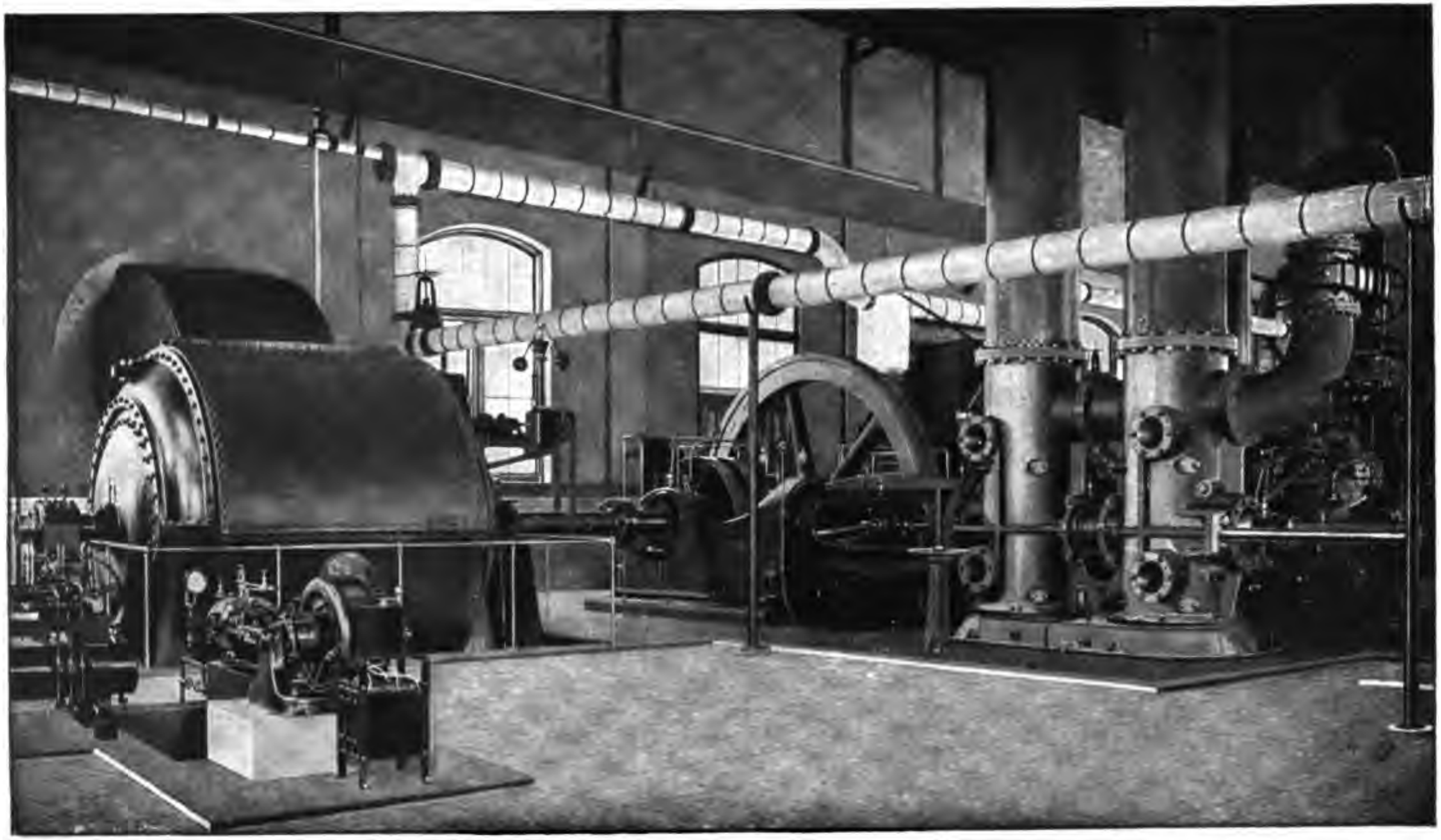
Fig. 3.

As seen in Fig. 3, the plunger P on the outward stroke draws water from the reservoir through pipe A and inlet valves E into the barrel B. On the inward stroke, the valve E will close and, since liquids are practically incompressible, the discharge valves D will open allowing the water to pass into the discharge chamber and out the pipe C. Fig. 3 represents a single-acting plunger pump, but the action is the same for a piston or for a double-acting pump except that there are two sets of valves. Although the force is exerted on the mass of liquid in a direction lengthwise of the barrel, yet, since pressure is transmitted equally in all directions, the valves will open promptly if located in any position with respect to the barrel.

The pressure against which the pump must act in forcing depends on the weight of the valves, the spring pressure, if any, used to close them quickly and the height to which the liquid is to be raised, or as it is technically termed, the "head," against which the pump is to work. For water, the pressure per square inch increases one pound for every 2.32 feet, or as previously stated one foot of head produces 0.433 pound pressure per square inch. There is no theoretical limit to the height to which a liquid may be raised by forcing, but the speed of starting and stopping the column of water, that is, the number of strokes per minute, must be decreased as the delivery pipe becomes longer, if water-hammer is to be avoided; this is true whether the length be horizontal or vertical. This hammer effect is due to the force necessary to overcome the inertia of the column of liquid and start or stop its motion. If not properly taken into account, either by using slow speeds or providing adequate air chambers to take up the shock, it will always produce destructive results. Wear on the outlet valves also limits the height to which water may be forced, as this wear increases with the head and makes it difficult to keep the valves tight.

Usually no special precautions are required up to pressures of 500 pounds per square inch; above this, care must be exercised up to the limit of 2,000 pounds, which is as high as will be required for lifts or presses except in special cases. Anything above this requires the greatest precautions, not only as to strength of parts, form of valves and packing of joints, but even with respect to quality of castings. Water will sometimes leak through what appears to be solid metal.

The pressure produced by water-hammer is taken advantage of in one special form of pump, **the hydraulic ram**, in which the



PUMPING STATION WITH RIEDLER DUPLEX PUMP,
Driven by Corliss Engine or by Direct Connected Water Turbine.

TABLE II.

Head in feet.	Pressure pounds per square inch.	Head in feet.	Pressure pounds per square inch.	Head in feet.	Pressure pounds per square inch.
1	.43	46	19.92	91	39.42
2	.86	47	20.35	92	39.85
3	1.30	48	20.79	93	40.28
4	1.73	49	21.22	94	40.72
5	2.16	50	21.65	95	41.15
6	2.59	51	22.09	96	41.58
7	3.03	52	22.52	97	42.01
8	3.46	53	22.95	98	42.45
9	3.89	54	23.39	99	42.88
10	4.33	55	23.82	100	43.31
11	4.76	56	24.26	101	43.75
12	5.20	57	24.69	102	44.18
13	5.63	58	25.12	103	44.61
14	6.06	59	25.55	104	45.05
15	6.49	60	25.99	105	45.48
16	6.92	61	26.42	106	45.91
17	7.36	62	26.85	107	46.34
18	7.79	63	27.29	108	46.78
19	8.22	64	27.72	109	47.21
20	8.66	65	28.15	110	47.64
21	9.09	66	28.58	111	48.08
22	9.53	67	29.02	112	48.51
23	9.96	68	29.45	113	48.94
24	10.39	69	29.88	114	49.38
25	10.82	70	30.32	115	49.81
26	11.26	71	30.75	116	50.24
27	11.69	72	31.18	117	50.68
28	12.12	73	31.62	118	51.11
29	12.55	74	32.05	119	51.54
30	12.99	75	32.48	120	51.98
31	13.42	76	32.92	121	52.41
32	13.86	77	33.35	122	52.84
33	14.29	78	33.78	123	53.28
34	14.72	79	34.21	124	53.71
35	15.16	80	34.65	125	54.15
36	15.59	81	35.08	126	54.58
37	16.02	82	35.52	127	55.01
38	16.45	83	35.95	128	55.44
39	16.89	84	36.39	129	55.88
40	17.32	85	36.82	130	56.31
41	17.75	86	37.25	131	56.74
42	18.19	87	37.68	132	57.18
43	18.62	88	38.12	133	57.61
44	19.05	89	38.55	134	58.04
45	19.49	90	38.98	135	58.48

force produced by the inertia of a large column of water suddenly checked is utilized to raise a small column against a high pressure. The working of the apparatus is seen from Fig. 4. Water flowing from pipe F into the chamber A will escape through valve V (closed in the figure) to discharge pipe D; but, as the velocity of flow increases, valve V will be lifted and closed checking the flow of water. The force thus developed will open valve V′ against the pressure of the spring and the head of water in the small pipe E, and some water will be forced into chamber B and up pipe E or into the air chamber C compressing the air. Water will pass into B until the force due to water hammer is so decreased that valve V′ closes and valve V opens once more, when water will again overflow and the process be repeated. The head against which the ram will work may be almost any amount, but the ratio of water raised to that wasted at the overflow will decrease as the head on valve V′ increases. In any case, the ratio will be small and the ram wasteful, so that its use is advisable only where the supply of water is almost unlimited, other power is difficult to procure, or the amount to be raised is very small. Table III gives the relation of efficiencies and ratio of lifting to forcing heads, which according to D. K. Clark, may fairly be expected with a ram of good design.

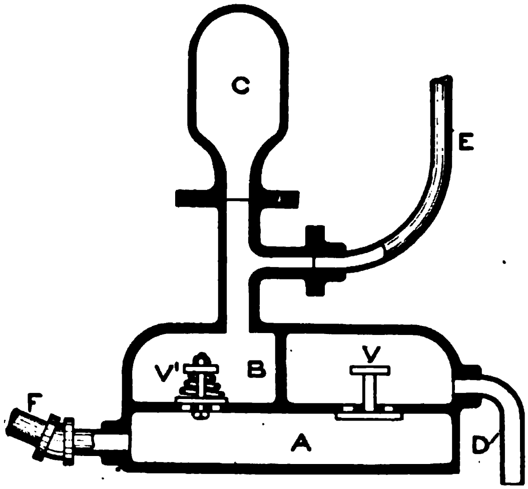


Fig. 4.

TABLE III.

Ratio of lift to fall	4	6	8	10	12	14	16	18	20	22
Efficiency per cent	72	61	52	44	37	31	25	19	14	9

The ram should be arranged with a drive pipe whose length is at least five times the fall, in order to give a long column of moving water and increase the water-hammer. The diameter of drive pipe is usually twice that of the lift pipe.

KINDS OF PUMPS.

In classifying pumps, the division may be made according to any one of several methods: As to the principles of action, the details of construction, the means of driving, or the purposes for which they are used. With respect to principle of action the classes may be:

1. Lifting,
2. Forcing,
3. Combinations.

With regard to the details of construction:

1. Jet,
2. Rotary,
3. Centrifugal,
4. Reciprocating.

With regard to uses, the division is difficult, since the same pump is frequently used in different places for different purposes. Such classes as tank pumps, boiler-feed pumps, air pumps and mine pumps are fairly definite, but the details vary so much with different makers that it is impossible to draw sharp distinctions.

From the discussion of the principles of action we know that practically all power pumps are of the third class, that is, work by a combined lifting action due to suction and forcing. But the second classification is yet to be considered.

The Jet Pump. The first variety, the jet pump, is perhaps best known in the form of the injector, but the principle is worthy of a much wider application than it now has. In the form of a steam or water ejector it may be used for lifting water by suction and raising it to a considerable height, while its simplicity and ease of application give it a large field of usefulness. A convenient form for an ejector is shown in Fig. 5. The propelling jet may be either steam or water under pressure entering through the valve A and flowing from nozzle D which should be made with a decreasing taper like a hose nozzle. If steam is used, it will be necessary to provide a small injection pipe with valve for supplying water to condense the jet and create a vacuum at the start, otherwise the steam will blow through the apparatus and will not lift the water in the suction pipe. In this case, the heat energy of the steam will be converted into velocity, and will be available for drawing and forcing the water so that it can be raised to a considerable height. The efficiency of a well-designed steam ejector should be nearly equal to that of an injector and will, like that, depend on the temperature of the water and the height of the suction and lift.

In pumping into a boiler, the efficiency of an injector is nearly 100 per cent because nearly all the heat energy which is lost in friction goes to heat up the feed water and is therefore utilized; but for other purposes, the injector or ejector is wasteful of heat and cannot be considered economical. Its convenience, however, is a strong point in its favor, and where the amount of water to be lifted is not large, or if it is to be used only occasionally, it will often pay to use it rather than to install a more elaborate and costly pump.

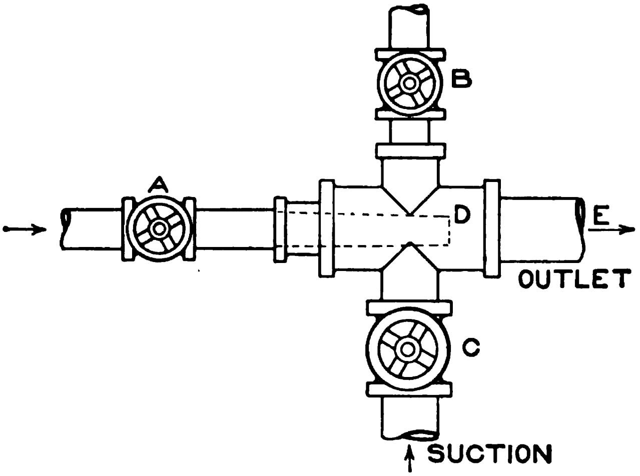


Fig. 5.

When water under pressure is used as driving power, it is not necessary to have the inlet at B. The flow of the water from the nozzle will carry along the air and create a vacuum at D which will draw the water up the pipe C. There is no heat energy available as in the case of steam, so that the water can furnish only such an amount of power as is due to its weight and the velocity with which it leaves the ejector nozzle; and, as the efficiency is low, not over one fourth of the energy furnished to the ejector at A will be available to move water at E. For a water injector it is well to proportion the pipes so that the head on A times the area of D equals twice the suction and lifting heads combined multiplied by the area of C; and to make the area of E equal to the area of D plus the area of C, if the apparatus is to be used as a water ejector. For steam, a $\frac{1}{2}$-inch pipe at A for a 2-inch at C and a $2\frac{1}{2}$-inch at E has been found to work well when the head on E was small, and if the water is under city pressure a $\frac{3}{8}$-inch pipe at B.

It is well to remember that the energy of a jet of either steam or water depends more on the velocity than on the weight flowing, since the energy available per second equals $\frac{W v^2}{2 g}$ where W is the weight flowing per second, v is the velocity in feet per second and g is the acceleration due to gravity; therefore, for good efficiency, it is best to use high pressure whenever possible.

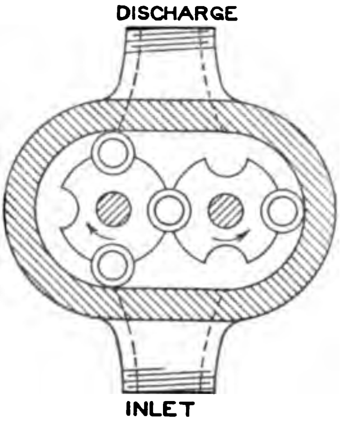


Fig. 6.

Rotary Pumps. These have been but little used because, as usually designed, their efficiency has been low. They are, however, useful in many cases and are well adapted for moving large volumes of liquid at low pressures. This pump moves the liquid by catching it between the lobes of the revolving propellers and the casing of the pump and sweeping it around from the suction pipe to the discharge; its return being prevented by the obstacle of the lobes and cylinders in the center. All are made on the same principle, the difference being in the design of the lobes and cylinders.

These are of many forms, but two will be sufficient to illustrate the principles. Fig. 6 is the Berrenberg; here the lobes are made by two pieces of tube fastened opposite each other. The tubes extend from the semi-cylindrical openings and make a joint with the other lobe by fitting in as shown. The axles are geared together outside the casing and are supported by bearings inside and outside, which are tapered to allow taking up wear and keeping the shafts true. The curves in Fig. 7 show the results from a series of tests made on a 2-inch pump by Mr. R. A. Hale and recorded in Barr's "Pumping Machinery." These show that the efficiency of the pump increases with the pressure against which it is working up to a certain point, and then falls off, while comparison of the curves at different speeds shows that the efficiency becomes less as the speed increases. This is apparent because the loss from friction is not doubled by doubling the head, while the amount of water will be the same per revolution, no matter what the pressure may be. The exception to this is due to slip which increases as the pressure becomes greater. Hence the useful work will increase up to that point where the loss due to leakage and friction more than balances the increase of work due to the increased head. The same reasoning applies to the speed, except that in this case the friction work will increase in proportion to the speed, so that the gain from the added work due to the increase of speed is not as great as from increase of head.

RELATION OF EFFICIENCY TO PRESSURE FOR ROTARY PUMPS.

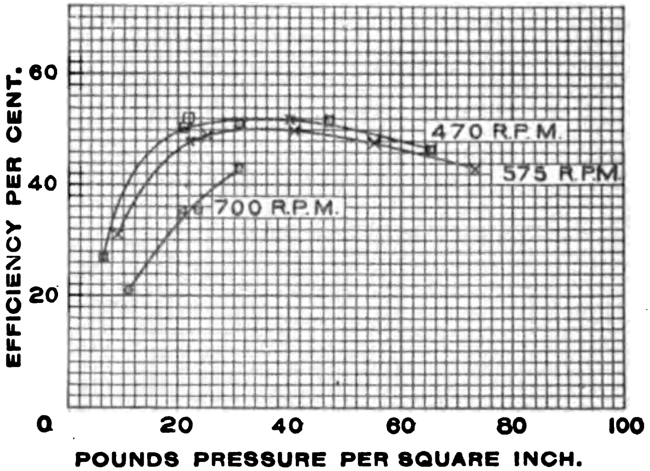


Fig. 7.

Another style of rotary pump is shown in Fig. 8 in which the two pistons are so designed that their bounding surfaces are always in contact, thus forming the joint. In this type it is important that the surfaces be of the correct shape, otherwise a pocket will be formed in which water will be caught. This water will be compressed as the volume of the pocket becomes smaller. As water is practically incompressible, it must either leak past the joint or burst the pump. In order to prevent this, a port is sometimes cut in the casing to allow the water to bypass and relieve the pressure. If, however, the outline of the lobes is made of the right shape, there will be no compression and the ports will not be necessary. It has been demonstrated that the curves to use for the outline of the lobes are the cycloids, as they can form the whole outline and will make a smooth surface which will not create pockets at any part of the revolution. The involute curve has sometimes been used, but it is not good as it does not return on itself, and must therefore be cut off by the arc of a circle on the end of the lobe, leaving corners which form compression pockets.

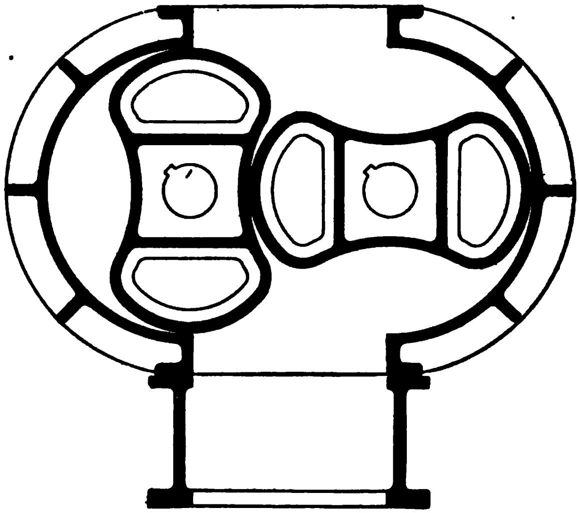


Fig. 8.

With respect to the effect of the speed on the volume of water pumped, the experiments above noted showed that, within the stated limits, the volume varies nearly as the speed, and that at the same speed, the volume is a little less at high pressure than at low, but does not fall off rapidly as the pressure increases. The data, however, is not complete enough to be conclusive.

The rotary pump is often used for fire purposes, and where portability is of more importance than efficiency, or if first cost is a great consideration. It can be run at high speed, and has a large capacity for the space occupied, but it is not suitable for large or permanent work except at very low heads.

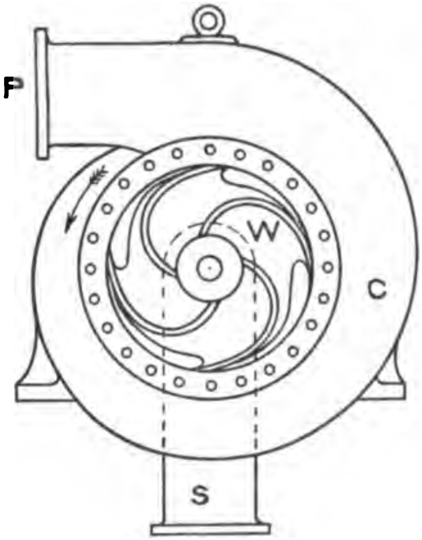


Fig. 9.

Centrifugal Pumps. These operate by the familiar force which throws the mud from the carriage wheel and the water from the lawn sprinkler. A wheel W (Fig. 9) with curved blades revolves in a casing C and, when the casing is filled with water, throws it to the outside thus creating a suction at the center and drawing up water through the suction pipe S. The velocity and whirling motion given to the water send it to the outside of the casing and out through a delivery pipe P. As seen in Fig. 9, the casing increases in size up to the outlet P, as the volume of the water to be carried will increase in that way. The blades are given a backward inclination at the wheel center so that water may enter without shock, and the angle which the blades make with the radius is increased towards the outer end so that the water gradually attains some rotary velocity and great radial velocity due to the wedging action as it passes outwards; on leaving the blades it has high speed. If the blades are radial at the outer end, the discharge will be greater for a given speed, but it will not be possible to reduce the speed much without stopping the action altogether; also the efficiency will be less.

Rankine states that the curve of the blade should be an involute of a circle, that is, a curve drawn by a point in a string as it is unrolled from the circle as shown in Fig. 10. The method of drawing the involute is as follows: Let BC be the inner bounding circle of the blades and B the point at which a vane is to start. Draw a radius of the circle through B and through the same point at an angle of 40 degrees with the tangent, another straight line BE. Draw a concentric circle tangent to BF, which is perpendicular to BE. If B were a point in a string, and this were unrolled from the circle, the point B would trace an involute BGH. For the first form of curve mentioned, this would be continued to the outer bounding circle where it should make an angle of about 15 degrees with the tangent, as LHM. For the Rankine blade, the reverse curve HKA should be drawn, usually the arc of a circle radial at the point H. In order to work without shock, the angle of the blade with the tangent at the inside end B should be such that its tangent equals the radial velocity of the water divided by the circumferential velocity of that point on the wheel; it is often made 40 degrees, which is found to be satisfactory for the average case. The angle at the outside should be 15 degrees if there is no diffusing chamber, or if the discharge pipe is not of increasing diameter, in order to gradually lessen the velocity of the water after it leaves the wheel. But this will necessitate a high rotary velocity of the blades, and it is sometimes better to use a diffusing chamber (or an increasing discharge pipe), and make the blades of the Rankine form, running the wheel at a lower speed unless the conditions make it necessary to be able to run at a given medium speed for a partial discharge. The casing should be shaped to give the water a gradually increasing velocity from the time that it enters the suction pipe until it reaches the circumference of the fan, and then a decreasing velocity until it leaves the discharge pipe. The spiral casing is often recommended because the centrifugal force given by the rotary motion is radial, not tangential, and the water must be

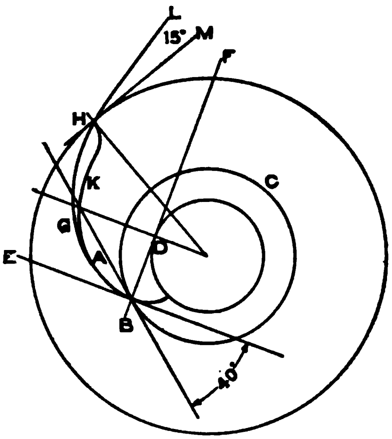


Fig. 10.

given time to change its direction or it will spend much of its energy in eddies and useless work.

Three styles of blades are shown in Figs. 11, 12 and 13. Fig. 11 is the form used for small sizes and for thick liquids; Fig. 12 is a hollow-arm type used in large pumps; it has the advantage that the water is thrown outward without any churning motion, and that there are no dead spaces. Fig. 13 is the style used for dredges and has the advantage that the sand is kept from grinding between the blades and the casing yet large openings are free for the passage of sand and mud.

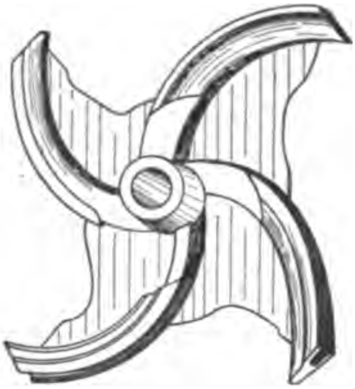


Fig. 11.

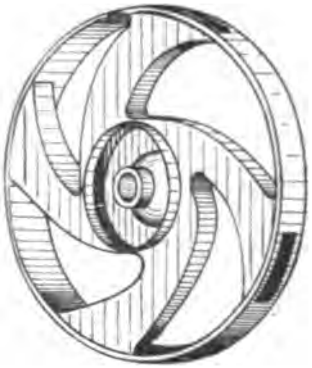


Fig. 12.

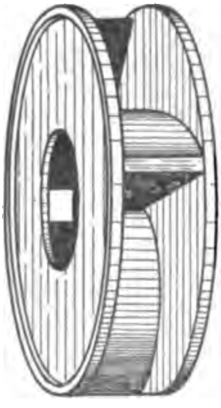


Fig. 13.

It is possible to raise water to any height up to 100 feet by increasing the speed of the pump, but it has been shown by experiment that this form is not suitable for lifts of over 25 feet, although up to 35 feet the efficiency is fairly good. As it is at these low lifts that the reciprocating pump is least efficient, this is clearly the field of the centrifugal pump.

The speed is increased as the height of the lift increases, but not in exact proportion, because the increase of speed for an increase of 10 feet in lift is greater at low than at high heads. Fig. 14 shows the relation between head and speed in two pumps of the same make, the No. 2 having a capacity of 100 gallons per minute at the speeds given and the No. 6 a capacity of 1,200 gallons. The point at which the curves cut the axis of speed shows that the No. 2 pump will fail to work at a speed below 250 r. p. m. and the No. 6 below 125 r. p. m. The pump may be set in either a vertical or a horizontal position and may have the suction on one side or both. However, it is better on both so that the forces due to lifting the water in the suction pipe are balanced and no end thrust produced. It is necessary to have some means of filling the pump casing with water, that is, "priming" it, as the suction will not start until the pump is so primed. If water under pressure is not available, it is necessary to install a steam injector to fill the casing, or provide a foot valve which will be tight enough to hold the water in the suction pipe at all times.

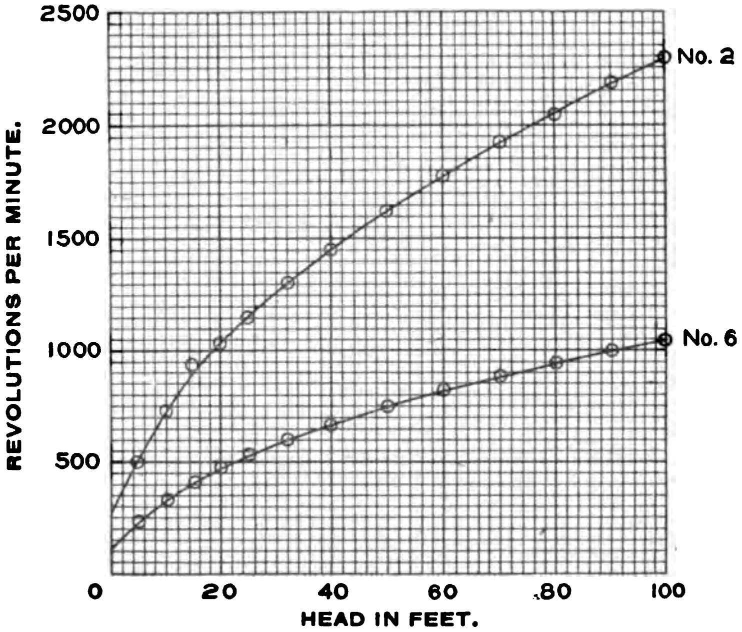


Fig. 14.

Makers recommend that a flap valve be installed at the top end of the discharge pipe also, to assist this action. Fig. 15 shows an arrangement which is satisfactory and also gives an idea of the shape and relative proportions of a standard centrifugal pump.

Lifting Pumps. The only pumps which work entirely by lifting are the old-fashioned chain devices in which a series of discs mounted on a chain are drawn, by a sprocket wheel, up through a tube bringing the water with them. In this form the motion is always in one direction, and there is a purely lifting action. But it is common to speak of the pump which delivers its stream by lifting, as a lifting pump even though it is filled by the action of suction. These are used largely in mines and for domestic purposes. The principle has been shown in Fig. 2. The disadvantage of this style is that the pump bucket must be not over 20 feet above the water and the rest of the lift must be made by the rise of the bucket, necessitating either a long rod, or pumping in relay. The greatest use of this class was in the Cornish mines in the time of Watt when the engines were placed at ground level, and the rods made of a length sufficient to reach to within 20 feet of the bottom of the shaft. In some cases these pump rods were as much as 480 feet long, and the design of the great pieces was a difficult problem. The walking-beam engine was used largely to drive the pumps, and the whole machine was a huge and wasteful affair.

At the present time, the long pump rod is used to some extent, but it is driven by a vertical cylinder set over the top of the mine shaft.

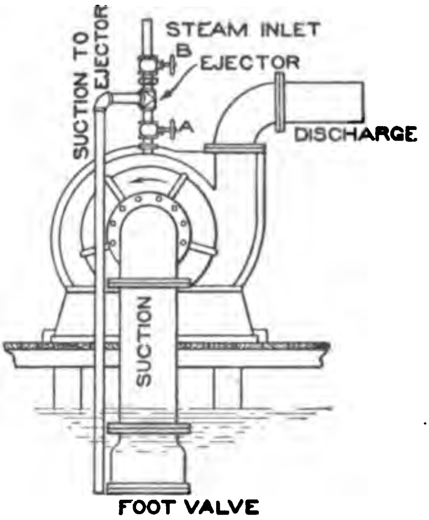


Fig. 15.

The size of the pump cylinder depends upon the volume of water to be raised and the speed of the pump. When heavy masses are to be moved, as is the case when long rods are used, the speed must, of course, be slow. For the great engines of Watt's time it was about 15 strokes per minute, and in modern pumps it is 25 to 35 strokes per minute. The cylinder must be made of such size that at this speed it will lift the required amount of water.

For example, if it is desired to raise 100 gallons per minute, and the pump is to run at 30 strokes, the volume to be raised is $100 \times 231 = 23{,}100$ cubic inches, and the volume per stroke is $23{,}100 \div 30 = 770$ cubic inches. A usual value for the stroke of pumps of this size is 16 inches which (if there were no allowance for leakage) would make the area of the cylinder $770 \div 16 = 48.13$ square inches. The allowance for leakage or slip should be, for a pump of this size and speed, about 3 per cent. This would, of course, increase with high speed and decrease with large size. Allowing 3 per cent slip, the area should be $48.13 \times 1.03 = 49.58$ square inches which is that for a circle of about $7\frac{15}{16}$ inches diameter, so that 8 inches would be used and the area would then be 50.26 square inches.

To find the force needed to lift this bucket, it is necessary to know the height to which the water is to be raised. Suppose this to be 200 feet; then the pressure per square inch will be $200 \times .433 = 86.60$ pounds and the force needed, $86.60 \times 50.26 = 4{,}352+$ pounds To this must be added the weight of the bucket and rod as both must be lifted at each stroke by the driving head at the top of the shaft. The rods are usually made of wood with the sections fastened together by iron couplings. The proper size for the above pump is 4 inches diameter, which for a rod 200 feet long would weigh, at 45 pounds per cubic foot, $45 \times \frac{\pi \times 4^2}{4 \times 144} \times 200 = 785$ pounds (about), and the bucket and fastenings will easily make this 1,000 pounds; the whole weight to be lifted will then be 5,352 pounds, which, at 30 strokes per minute, will call for the expenditure of

$$\frac{5{,}352 \times 30}{33{,}000} \times \frac{16}{12} = 6.5 \text{ H. P. (nearly)}$$

This would be the power necessary to run the pump aside from the friction of the machinery.

Although, at the present time, high lifts are largely made by using a forcing pump at the bottom of the shaft, this was not known at the time of the introduction of the pumping engine in the Cornish mines, and when it was necessary to raise water from a depth greater than could well be handled in one lift, it was accomplished by using pumps in relays. One engine would be installed at a depth of some 240 feet from the bottom of the mine, with its pump cylinder within 20 feet of the bottom, and would raise the water into a reservoir from which it would be taken by another pump and so on. Instances are recorded in which pumps raised 4,614,800 gallons per 24 hours, having pistons 24 inches in diameter and a stroke of 10 feet.

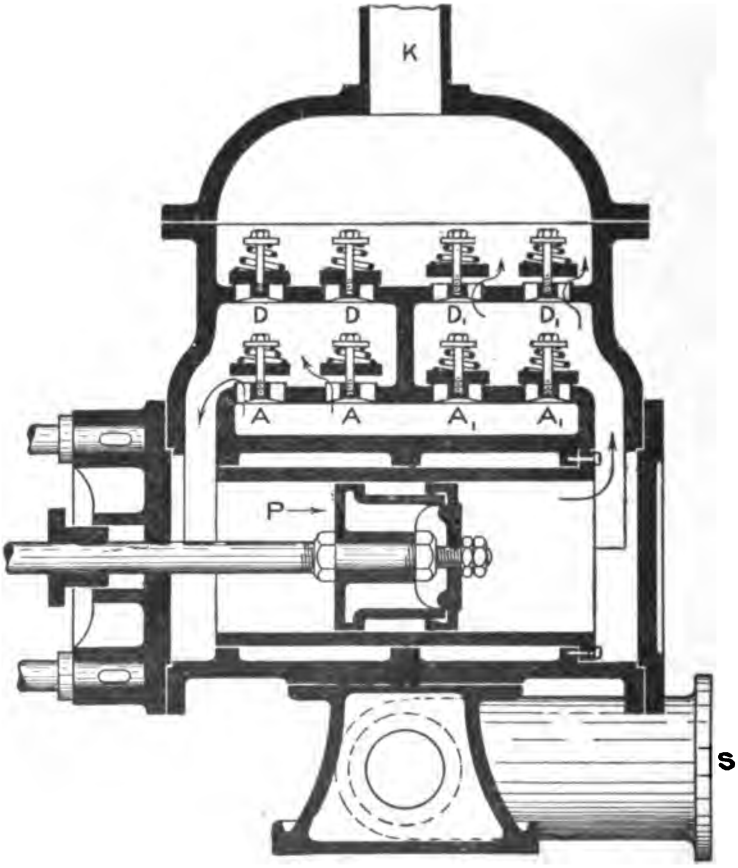


Fig. 16.

Forcing Pumps. For raising water to a height greater than can be conveniently overcome by lifting, or when the discharge is desired at a distance from the pump, it is better to use the forcing type. The simple principle of this type is shown in Fig. 3. There are, however, many modifications in common use. It is usual, if the lift to the pump be great, to install a foot valve at the bottom of the suction pipe, as well as to use admission valves to the pump chamber so that the suction pipe will be kept filled at all times, and the tendency for the water to drop out of the pump chamber, while the suction valve is closing, be removed.

The most common form of forcing pump is the double-acting piston type as seen in Fig. 16. As the piston P moves to the right, it will create a suction in the left-hand end of the cylinder, drawing in water through the valves AA and the suction pipe S. At the same time pressure will be exerted in the right-hand end of the cylinder and water will be forced out of valves D_1D_1, into the discharge chamber and up the pipe K. On the reverse stroke, water will be drawn in at valves A_1A_1 and forced out of DD. The piston must, of course, be packed water tight.

As in the case of lifting pumps, the volume to be raised and the speed at which the pump can be run determine the size of the piston; but the speed is usually greater than in the lifting type. The pump must not be placed over 20 feet above the level of the water which it is to raise, hence for a high head the greater part of the lift must be obtained by forcing against the head above the pump. It is a common rule that the travel of a pump piston should not exceed 100 feet per minute, and, taking the same problem as for the lifting pump this gives, $\frac{23{,}100}{100 \times 12} = 19.25$ square inches as the area of the piston. At the increased speed, the slip will be greater, hence it will be necessary to allow 5 per cent or an area of $19.25 \times 1.05 = 20.21$ square inches corresponding to a diameter of $5\frac{1}{16}$ inches.

The head would be the same, hence the total pressure per square inch would be the same as for the lifting pump, but the power would be increased. The total force on the piston will be $20.21 \times 86.60 = 1{,}750.1$ pounds. At 100 feet per minute this will require $\frac{1{,}750.1 \times 100}{33{,}000} = 5.30$ horsepower, which is less than before because the weight of the pump rod is not lifted at each stroke. If the rod be counterbalanced by a weight at the opposite end of a walking beam, the lifting pump will have a slight advantage owing to the smaller slip.

The pump shown in Fig. 16 is double acting, that is, it works on both strokes, but pumps, especially of the vertical type, are often made single acting to suck on the up stroke and force on the down; these have only half the capacity for the same size as the double acting.

The great difficulty with the piston is in keeping the packing tight or of knowing when it is leaking; it is also difficult to repack as the pump must be dismantled in order to get at the piston. For this reason the plunger pump is often preferred and is used either single acting with stuffing box around the plunger as in Fig. 3, double acting with two plungers as in Fig. 17, or double acting with one plunger as in Fig. 18. The advantage of this last over the piston is that for high pressures it is more easily kept tight, and for gritty liquids the wear is taken by the bushing which is more easily and cheaply replaced than a lining to the cylinder. The piston pump is more compact; but the plunger

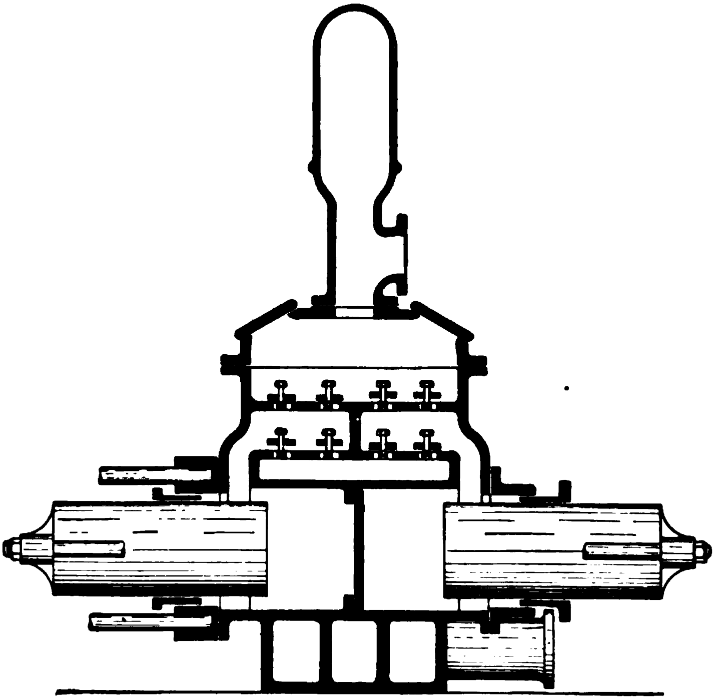
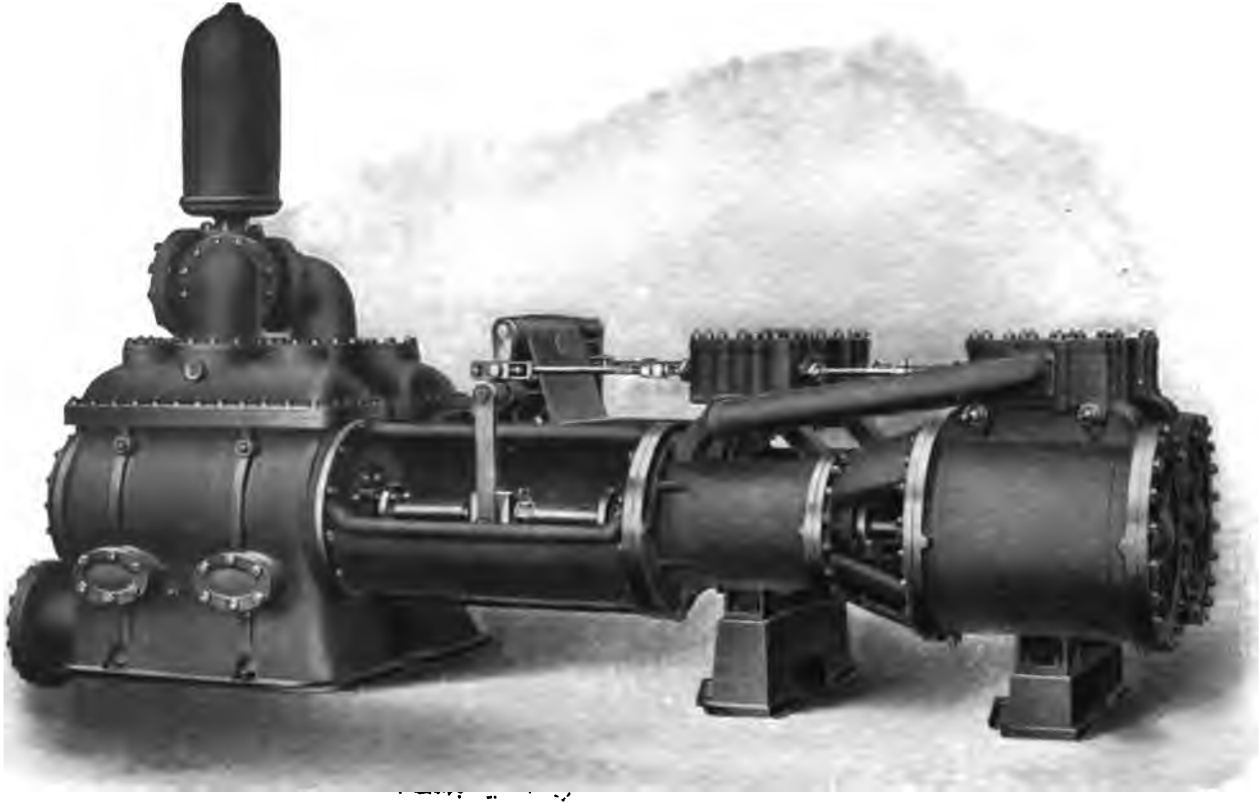


Fig. 17.



WORTHINGTON COMPOUND PISTON PUMP, WATER-WORKS PATTERN does not require a bored cylinder so that the first cost is not materially different.

The two-plunger pump with outside yoke is still larger than the single-plunger double-acting type but it can be easily repacked, can be kept tight against high pressures and any leaking is instantly detected. For these reasons, it is a favorite. The double-acting pump, not only has greater capacity, as above mentioned, but, on account of its continuous action produces a much steadier flow than the single-acting type, so that it is generally preferred.

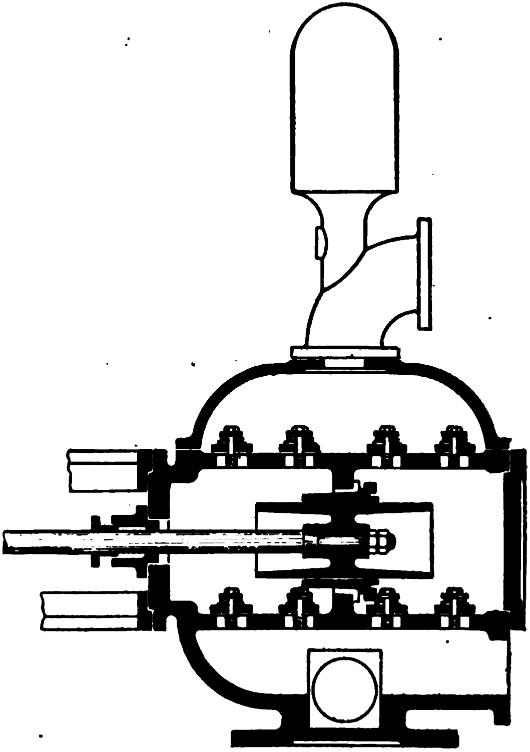


Fig. 18.

A modified form of Worcester's Engine, Fig. 1, is seen in the pulsometer or hydrotrophe, Fig. 19, which is still used to a considerable extent where extreme simplicity is desired. The ball B rolls so as to close first A′ and then A; with the ball in the position shown and the chamber D filled with steam, water will be drawn up through F by the vacuum caused by condensation of the steam. Meantime steam will enter D′ through A and force the water out through the check valve G. This will continue until the steam in D is all condensed, when the rush of the water, due to its inertia, will drive the ball valve B over against port A, thus reversing the action. It is entirely automatic and, as the machine is the acme of simplicity and may readily be made with large openings, it will handle dirty liquids with a minimum of attention. Its usual heat efficiency is 1.2 to 1.5 per cent, but this is not as objectionable as. it seems, since small reciprocating steam pumps often do little or no better. C is a suction air chamber connected with the opening E to equalize the flow of water.

VALVES.

Pump valves must be so made that they will open and close quickly and fully with little friction, so that there shall be no obstruction to the passage of the water, yet little leakage of water past the valves after the reversal of motion of the piston and before the valves have seated. Quick opening requires a light valve with slight motion, but quick closing demands either a heavy valve or a spring to hasten its action. In older pumps it was the custom to use one large valve with a lift sufficient to give the required passage; the valve was then heavy enough to close of its own weight, but was clumsy and hard to fit to a seating. In modern practice the required area is divided among several small valves so that each one is easily and cheaply renewed in case of accident or wear, and quick closing is obtained by the use of springs on the spindles of the valves.

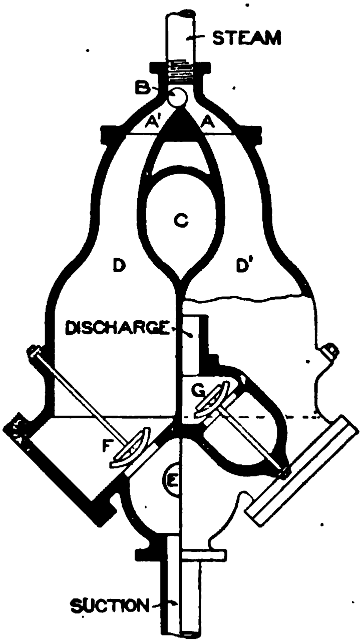


Fig. 19.

The natural classification of valves is by the motion and by the method of securing a tight joint; according to the motion there would be division into flexible, hinged, poppet and ball; according to the seating, into cushioned and ground.

The Flexible valve is the oldest type, and is, in some ways, the simplest. It is generally made of rubber, as this material gives the greatest flexibility with the best seating. The valve is made as a solid disc, the opening being obtained by the flexibility of the valve itself, and is usually of the style shown in Fig. 20. The seat is made with radial grids to take the pressure of the water, the openings between the bars being made equal to the thickness of the disc; this thickness depends in turn on the diameter of the disc, and is often made as given in the following table:

Diam. of disc. Inches.	Thickness of disc. Inches.	Diam. of hole in center. Inches.
2	$\frac{3}{8}$	$\frac{1}{2}$
$2\frac{1}{2}$	$\frac{7}{16}$	$\frac{1}{2}$
3	$\frac{1}{2}$	$\frac{9}{16}$
$3\frac{1}{2}$	$\frac{5}{8}$	$\frac{11}{16}$
4	$\frac{5}{8}$	$\frac{11}{16}$
$4\frac{1}{2}$	$\frac{3}{4}$	$\frac{13}{16}$
5	$\frac{3}{4}$	$\frac{13}{16}$

The valves of this type most commonly used are the 3-inch and the $4\frac{1}{2}$-inch as the larger sizes are not found durable. The 3-inch is the best for ordinary use. The hole in the center is made $\frac{1}{16}$ to $\frac{1}{8}$ inch larger than the spindle to allow of free motion; to avoid sudden bending of the rubber, the guard is placed to give the disc a chance to rise about $\frac{3}{16}$ inch. When it has risen so far, it bends around the guard and thus gives a free opening for the passage of the water. The guard should have a diameter equal to that of the disc less twice the thickness of the disc, and the radius of curvature should be equal to the diameter of the disc. This type of valve is not well suited for high pressures, 250 feet head being about the limit for the ordinary style. If, however, there are vanes arranged below the disc so that a rotary motion is given to the valve, it is found that there is no difficulty in keeping the valve tight for heads up to 490 feet. For these pressures, it is not uncommon to use discs of a thickness up to an inch, but only for large sizes. The lap of the valve over the edge of the seat should be from $\frac{1}{4}$ to $\frac{1}{2}$ inch according to the thickness of the disc and the pressure to be carried.

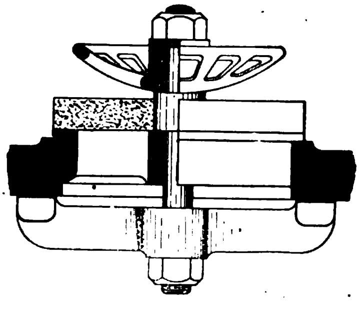


Fig. 20.

Leather valves have been used to some extent in the form of a simple flap fastened at one side and free to rise at the other; this makes the simplest form of hinged valve and, when weighted with a metal disc on either side, has been much used for small hand pumps. It is not suitable for severe service.

The Hinged valve, as its name implies, is made to swing about a pivot at one side, but should be designed to lift from its seat before it begins to swing. The disc is made of metal but often with a facing of leather or rubber for slow speeds; for high speeds these linings will not stand the wear. The lift should be about $\frac{1}{8}$ to $\frac{1}{4}$ inch before the valve begins to turn about the fulcrum. The angle to which the valve lifts is usually made 30 degrees, and should never be over 60 degrees, in order to ensure quick opening and closing. The motion is limited by a stop on the seat or on the disc of the valve as seen in Fig. 21. The hinged valve is slow in action and is not suitable for thick liquors or high pressures. In designing, the area allowed should be as follows:

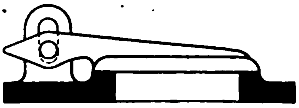


Fig. 21.

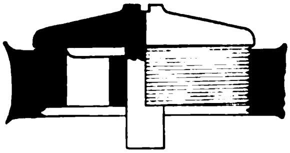


Fig. 23.

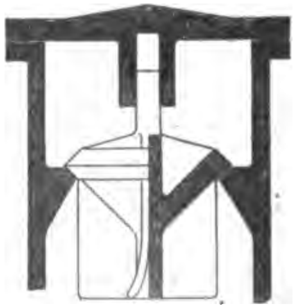


Fig. 22.

Piston speed, feet per minute............	100	125	150	175	200
Area of waterway, per cent of plunger area.	40	50	60	75	100

For higher speeds or for high pressures, it is best to use some form of positively driven valve, and, if such liquors as sewage or syrups are to be handled, the areas found from the above table should be increased about 50 per cent.

The most common form of valve in use at present is the **poppet** which rises straight up from the seat and gives a free opening with a small motion. The seat is made either flat or at an angle of 45 degrees as the designer may choose, the **beveled** seat being more difficult to construct but retarding the flow of water less than the flat seat which necessitates a double turn in the current. The width of the beat or edge of the seat against which the valve rests is made $\frac{1}{4}$ inch for large valves and $\frac{1}{16}$ inch for small ones. The flow of the water through the valves should be at a velocity of not over 200 feet per minute, and the lift must be so proportioned that the passage by the valve will be as large as the area of the valve opening. A valve with flat seat need not have over $\frac{1}{4}$ inch lap even for heavy pressures, but the joint must be carefully made to avoid leakage.

A common form of **mitre valve** is shown in Fig. 22 and of **flat seat** in Fig. 23. The vanes on the bottom of the disc which keep the valve centered when it rises are often given an inclination of $\frac{1}{8}$ inch in 6 inches so that the rush of the water will give the valve a rotary motion, removing all dirt and keeping the wear on the beat even. For the flat beat valves this is not always possible as they are closed by a spring and the friction prevents them from revolving.

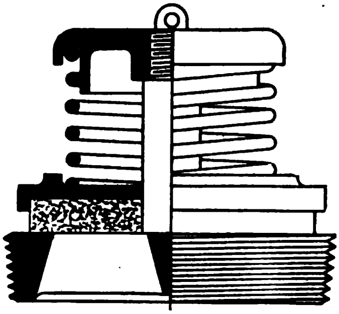


Fig. 24.

In order to give an area by the valve equal to the area of the seat, the lift must be one quarter the diameter of the seat opening. For heavy pressures this would make too large a lift, as the slip on closing would be excessive; for this reason the lift is made not over $\frac{1}{4}$ inch and the number of valves increased as may be necessary. The seat is usually made separate from the valve deck and screwed into place; when this construction is used, the guide for a valve (if there is one) should be made a part of the seat so that the valve is self contained. The guides may be in the form of wings on the bottom of the valve disc, a spindle running through a hole in the seat or a spindle projecting upward through the valve; the last being most common.

The discs may be of metal or hard rubber. For hot liquids metal must be used, but for cold liquids hard rubber or composition is more common, usually with a metal cap to keep the disc in shape. The spring used for closing the valve should be cylindrical rather than conical, as the strain on the latter form comes largely at the small end of the spring and the wire is likely to break there. A design to be recommended is that of Fig. 24 which is described by Barr; see "Pumping Machinery," p. 57.

Double beat valves and those with multiple ports are not usually necessary but are sometimes used for large work.

Ball valves are often used for low pressures. For these the diameter of the ball is made 1⅓ times that of the beat and a cage is provided so that the ball shall always be kept central. The ball is of gun metal usually with a rubber casing or, for cold water, sometimes wholly of rubber. They are common in pumps used for mines and oil wells and often in connection with cup leather packings for the piston. If the water is gritty or has impurities, the valve seat should be raised above the deck so that there will be room for the sand to settle below the seat, as seen in Fig. 25.

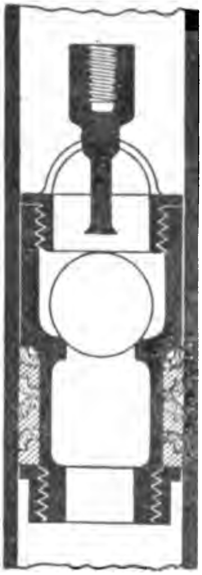


Fig. 25.

In all valves which have metal for both discs and seats, it is important that they should both be of the same material, as otherwise there is likely to be galvanic action which will cause rapid deterioration. Usually, however, it is better to use the valve with a separate disc of composition as this avoids all possibility of electrical action and makes the refacing of the valve unnecessary; the replacing of the old disc by a new one being all that is needed when the valve becomes worn.

For valves which are to be returned by a spring, it is best to use five coils in the spring and to have the wire of spring brass of the size given in the following table:

Diameter of valve inches..............	2	2½	3	3½	4	4½
No. of wire B & S....................	12	12	10	10	8	8

In this style, it is well to use for the guiding spindle a stud screwed solidly into the valve seat or made a part of the casting, and fasten the valve by a nut at the top of the stud, secured by a pin. This is usually better than to use a stud screwed into the valve seat. If the latter is used, it is likely to work loose and may cause much damage before the trouble is discovered; it is also more difficult to remove when repairing the valve.

The material for the valve depends on the liquid to be handled. For acids or liquors of that nature, it is best to use wood, although it is not durable, yet it will last better than any form of metal; for salt, petroleum products, and strong alkalies, gun metal is the best; for all ordinary uses, however, cast iron is good enough, and if used with composition discs there will be no trouble from rusting fast even if the pump is idle for considerable periods.

With respect to valves for large pumping engines, they are best made of double beat type, and usually of the "Cornish" form. For this a design used by Mr. A. F. Nagle has proven very satisfactory; it is shown in Fig. 26. The features embodied in this design are a small width of beat to avoid any possibility of sticking and to get rid of surface on which the pressure of the water cannot act to open the valve; a diameter large in comparison with the lift so that the velocity with which the water comes to the opening shall be small; a shape which avoids all sharp curves and changes of direction and all possibility of air pockets. The valve has been found to embody all these qualities, to work noiselessly even at high speeds, and to follow the motion of the piston almost exactly.

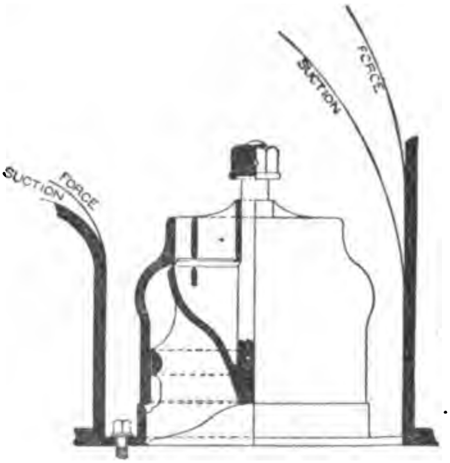


Fig. 26.

DESIGN AND CONSTRUCTION.

The essential parts of a reciprocating pump are inlet valves, a cylinder, a piston or bucket and outlet valves. The purpose of each is obvious. However, in order to get the best results certain points must be considered in the arrangement and construction of these parts. Some of these have already been mentioned in the description of valves. Admission and outlet valves are usually of the same construction but sometimes differ in size, the outlet valves being smaller and more numerous so that the lift may be less and the slip under high pressures reduced. Valve decks are usually arranged, one for the inlet and one for the outlet valves; into these decks are screwed or bolted the valve seats, as already shown in Figs. 20—24, the valves extending above into a valve chamber as shown in Fig. 16. The valves should have space enough between them and between the outside valves and the walls of the valve chamber so that the area for the passage of the water shall equal that through the valve openings. Generally, considerations of strength of the valve deck also demand as wide spacing as does the necessary water way.

The design of the **cylinder** depends entirely on the kind of pump; the considerations involved having already been mentioned. For a small and compact machine, the piston type is the best and this calls for a smoothly-bored barrel. In order that this may be easily renewed, it is usual to make the bored portion as a lining which fits an outer casting so that when worn by grit or long use, the lining may be renewed without remaking the whole water end. Such a construction is shown in Fig. 16.

If there is much sand or dirt in the water, it is usually better to use some form of plunger pump, as the round-turned plunger is much cheaper to renew than the bored-barrel lining; also the cylinder will be cheaper as there is little boring to be performed. The difficulty of an inside packed plunger has been mentioned and also the alternative of a double plunger, outside packed. (See Figs. 17 and 18). In spite of the difficulty of ascertaining and stopping leakage past the piston, and the expense of boring and reboring or relining the cylinder, the piston pump is used, except for high pressure, more than any other form on account of its compactness.

The bucket is rarely used except in vertical pumps and then almost entirely for mine or well purposes. It is of the form of Fig. 25 with possibly the clack or hinged valve in place of the ball. The outside of the bucket is packed by cup leathers or by hemp packing, the same as a piston, or it may simply have grooves turned in the outside to form a water packing. The barrel is often simply a brass pipe, bored to make a fit, and screwed to the end of the vertical pipe so that it may be easily renewed. In the case of deep wells, the bucket should be made with a long bearing surface so that it may the more easily keep a water-tight joint and may act as a steadier to keep the pump rod in line.

The **piston rod** for a pump is made much larger than considerations of strength would seem to demand, in order that there may be no vibration or deflection, as these would make it almost impossible to keep the stuffing box from leaking. Theoretically the rod for a long-stroke pump should be larger than that for a short-stroke of the same diameter; but, actually, the margin of strength is so great that the same size rod will answer for both. Allowance must be made in all cases for the reduction in diameter necessary to fasten the piston or plunger to the rod. This may be done by using a tapered end fitting a tapered hole in the body of the piston or by turning down the end of the rod and letting the piston bear against a shoulder; the latter method is the better. Usually a nut holds the piston in place, the nut being in turn secured by a lock nut or split pin through the end of the rod. Various modifications of these methods are of course used according to the fancy of the designer.

The diameters of piston and corresponding rod for a rod of cold rolled steel, tensile strength 65,000 pounds, a factor of safety of 10 and a pressure of 150 pounds are given in Table IV.

TABLE IV.

Piston diam. Inches.	Rod diam. Inches.	Diam. of rod end, Inches.
4	1⅜	1⅛
6	1⅝	1⅜
8	1⅞	1½
10	2¼	1¾
12	2⅜	1¾
14	2½	2

The **water end** of the pump is necessarily long, as the piston must be of such length that it can be packed, and usually with some kind of soft packing such as flax, hemp or leather. Fig. 27 shows the detail of a piston designed for use with square flax packing; the body is fastened to the piston rod by the nut A, the packing laid in place, and the follower forced up by the nut B which is, in turn, secured in place by the lock nut C. For large sizes, the design is the same except that the follower is set up by a number of nuts near the edge screwed onto stud bolts.

The leather-packed piston is made as shown in Fig. 28. The packing is made by pressing sheets of wet leather into iron forms and allowing them to dry there, after which the edges are finished, The radius of the corner must be large, so that the leather will not be injured in the pressing process, about $\frac{1}{2}$ to $\frac{3}{4}$ inch being sufficient. The diameter of the disc from which the cup is formed need not be more than $1\frac{1}{4}$ to $1\frac{3}{8}$ inch greater than that of the piston, as there is no object in having the lip of the cup longer than just sufficient to make a tight joint.

Less common forms of piston packing are the metal ring sprung into place and the series of grooves. For the former, the ring should be made about $\frac{1}{8}$ inch larger than the piston for a 12-inch piston; other sizes in proportion. For the groove packing, the grooves are best made about $\frac{3}{8}$ inch deep by $\frac{1}{4}$ inch wide. If soft packing is used, the length of a piece cut for a ring should be somewhat less than that needed to have the ends meet when placed around the piston body in order to allow for the swelling of the packing as it becomes wet. One rule is to make the space between ends equal to the width of the packing.

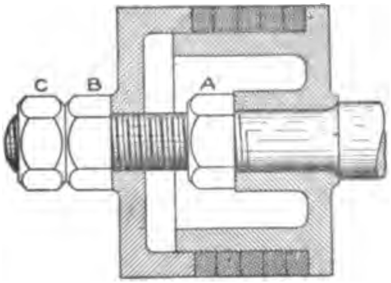


Fig. 27.

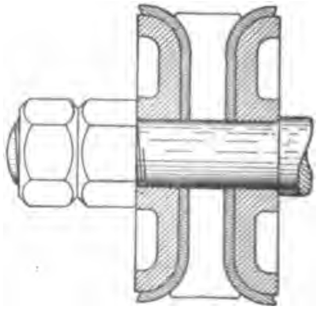


Fig. 28.

For a plunger, a hollow casting is generally used; the only requirement being that the thickness shall be sufficient to withstand the pressure against which the pump acts. The length must, of course, be equal to the stroke plus the length of the stuffing box and an amount sufficient to ensure that the head of the plunger will never strike the gland. A plunger is sometimes used with an open end and the partition set back into the interior of the plunger, but this is not as strong a construction as the closed-end type and is more expensive to make.

With single-cylinder pumps it is customary to use an air chamber on the discharge to steady the flow of the water and prevent shock, but for those having several cylinders this is not considered necessary. Even if the motion is slow, and the pressure against which the pump works is great, it is better to use the air chamber even with multiple cylinders, and it is often advisable for fast running pumps or those with a long suction pipe to use an air chamber on the suction side as well. The volume of the discharge chamber should be three to four times that of the piston or plunger displacement per stroke, and for the suction chamber twice such displacement if the pump is single. For duplex, ⅓ to ½ less will answer. The neck of the chamber should be long and narrow so that the air will not easily escape, and the flow of water into and out of the chamber will be somewhat retarded. For small sizes the chamber is made of copper in a standard form as seen in Fig. 29, the dimensions being about as follows:

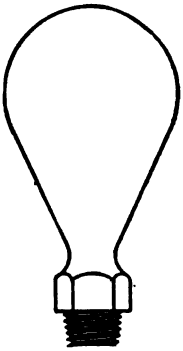


Fig. 29.

Diam. inches.	Height, inches.	Pipe-tap thread, inches.
6	10	1
8	14	1¼
9	15	1½
10	16	2

For larger sizes a casting is used with straight sides and a hemispherical top as seen on Fig. 17. For this style a common proportion is to make the height three times the diameter and the diameter of the neck one-third that of the chamber proper.

The air chamber on the discharge should be placed at the highest point of the valve chest and above the delivery opening so that the air will not tend to slip out with the water. Even then, the air will be gradually absorbed by the water, and provision must be made for renewing the air cushion either when the pump is not in use by admitting air through a cock and allowing some water to escape from the valve chest, or continuously by an automatic pump.

The suction air chamber should be so placed that the stream of water flowing to the pump may cushion against the air in it without changing its direction abruptly. Two positions which fulfill this condition are shown in Fig. 30. If it is impossible to place the suction chamber in such a position, the capacity should be increased considerably, the amount depending on the speed at which the pump is to be run.

In designing the valve chest, flat surfaces should be avoided, as they are not adapted to resist pressure. A rounded or oval surface gives greater strength and will more easily conform to the outline presented by the valve studs and the top of the chest. A good design is shown in Fig. 16.

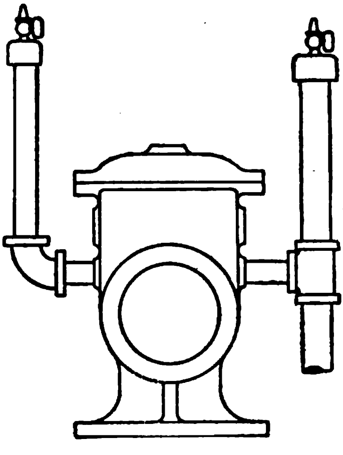


Fig. 30.

Clearance is the name given to the space at the end of the cylinder into which the piston or plunger does not travel. It is not, as is the case with the steam engine, a source of loss, hence it is advisable to provide a generous amount so that there will be a place for grit or foreign matter to settle, and no possible chance of the piston striking the cylinder head.

The ports or valve openings and the passages from the valve chests to the cylinder should be large enough so that the velocity of the water when the pump is working at its greatest speed will be not over 300 feet per minute. The ports should be short and direct to avoid friction, and if possible should be so arranged that the water passes into, through and away from the pump without changing the direction of its motion. This last condition usually conflicts with the necessity of placing the valves so that they are readily accessible and is, therefore, disregarded by many makers.

The stuffing-box for rods 1 inch or less in diameter is made with a cap to screw over the end of the box and forces in the gland as in Fig. 31. For larger sizes the gland is forced in by nuts screwed on stud bolts which pass through a flange on the gland. See Fig. 32. The gland and box may be bushed with brass as shown, or made of solid iron. The box should be made deep enough to allow four rings of packing and should preferably have the bottom and the end of the gland chamfered in order to force the packing against the rod and make the joint more secure.

The frame or body of the pump is to hold all parts together and keep them rigidly in position. With this end in view, it pays to use enough iron to make the frame massive, yet it should be so distributed as to be of the most use. The standards or legs should be of good length so that the pump may stand well away from the foundation and be accessible. The feet should have a large surface to secure stability and good bearing. For pumps which are to work against high pressures, round chambers connected by round passageways should be used for the valve chambers. The valve chambers should be so arranged that the valves or seats may be easily removed and replaced, a convenient arrangement being that of Fig. 16 where the inlet valves are directly under the outlet valves, hence can be readily examined Details of such an arrangement as used in the Cameron and Davidson pumps are seen in Figs. 33 and 34 respectively.

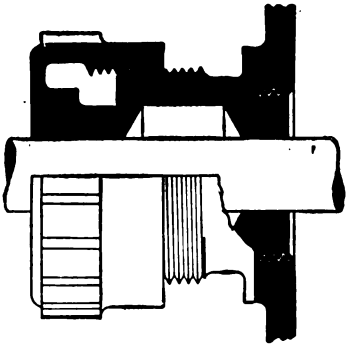


Fig. 31.

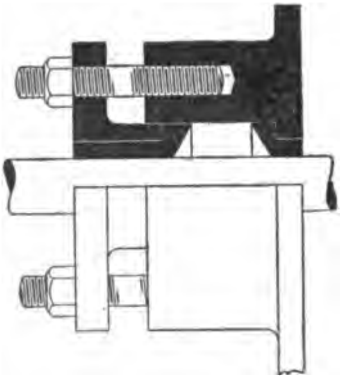


Fig. 32.

The arrangement of the parts of a pump is the test of a designer's skill. Air pockets must be avoided in the cylinder or clearance spaces, as otherwise the pump will waste much of its effort on the air thus caught. The passage through the cylinder must be so designed that the water leaves at the highest point of the chamber and all recesses in which air might be caught must be carefully avoided. Air pockets in the discharge valve chamber are not serious, but in the inlet valve chamber they will result in lost motion.

Almost all small horizontal pumps are arranged with both the inlet and discharge valve chambers above the cylinder, as this makes all valves readily accessible, and gives a compact water end. Unquestionably smoother action and a more efficient pump would be obtained if the admission valves were placed below the cylinder so that the water need not reverse its direction of flow while passing through the pump; but it is a case for judicious compromise, and the common arrangement gives good service so long as the speed of working is slow. Nevertheless, it is well to keep the better arrangement in mind, in which the suction enters at the bottom of the pump and the discharge leaves at the top.

The suction pipe usually comes in at one side, and the discharge leaves directly above it on the same side, as shown in Fig. 35, or the side opposite. Fig. 35 is the cross section of a single cylinder pump.

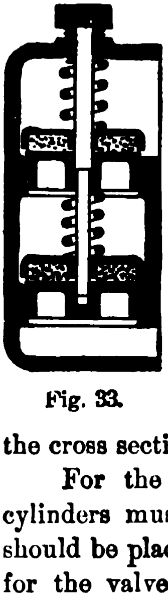


Fig. 33.

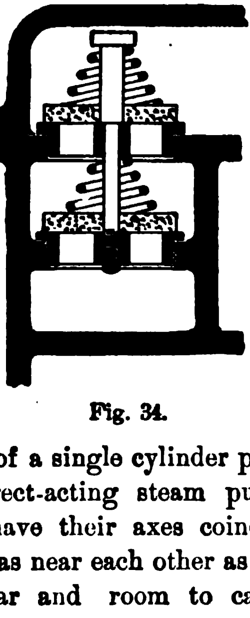


Fig. 34.

For the direct-acting steam pump, the steam and water cylinders must have their axes coincident. The two cylinders should be placed as near each other as is consistent with provision for the valve gear and room to care for the stuffing boxes. Compactness is desirable, but convenience in caring for the pump and in making repairs is much more essential.

At the lowest point on each end of each cylinder provide a drain cock by means of which all water may be drawn off when the pump is shut down in order to prevent rusting or pitting and to avoid danger of freezing in cold weather.

The size of the piston or plunger and length of stroke are determined by the capacity desired. The allowable speed of piston is often stated as 100 feet per minute, but it is better limited to 60 double strokes per minute, as the determining factor is really the time needed for reversal and changes of valves at the end of the stroke. For small pumps the limit might be made 100 double strokes per minute, but the smaller number is better. At 60 double strokes or 120 single strokes per minute the piston speed in feet per minute will be:

$$\frac{l \times 120}{12} = 10\,l$$

where l is the stroke in inches; and for 100 double strokes the speed will be $\frac{10}{6}$ times the above value or 16.7 l. For pumps of over 6-inch stroke this higher limit is not advisable.

Table V gives the capacities of various sizes of pump cylinders at 60 double strokes per minute, without allowance for slip.

From Table V can be seen directly the capacity of any size pump at a speed of 60 strokes per minute, or the diameter and corresponding length of stroke needed to move any given number of gallons per minute. To find the capacity at any other speed, divide the given capacity by 60 and multiply by the number of double strokes at which the pump is to run. Thus to find the capacity of a pump 3 inches diameter by 4 inches stroke at 80 strokes; the capacity at 60 strokes is 14.7 and at 80 it will be

$$\frac{14.7}{60} \times 80 = 19.6 \text{ gallons per minute.}$$

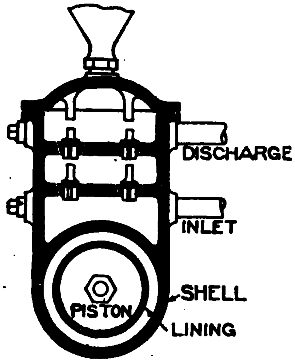


Fig. 35.

The ratio of stroke to diameter may be chosen by the designer, the value in standard designs running from 2:1 to 1:1.25, the more common being 3: 2 and 1: 1.

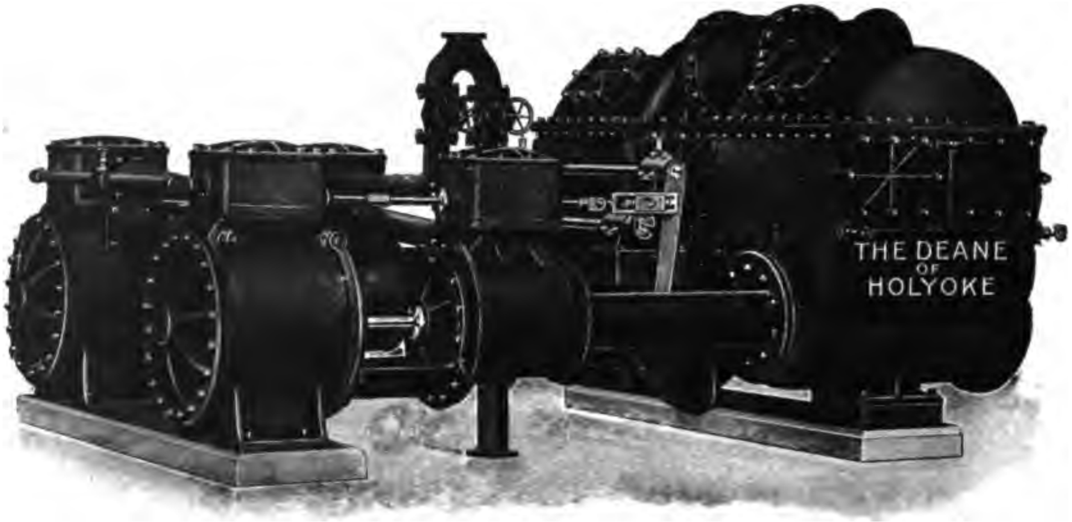
The first value to be found is the amount of water to be handled per minute, and this will usually be determined by the work for which the pump is to be used. The allowance for "slip," that is, the loss due to slow closing of the valves and leakage must then be added before the size of the water end can be settled.

The slip depends on the style of valve used, the size of pump

TABLE V.

CAPACITY OF PUMPS IN GALLONS PER MINUTE AT 60 DOUBLE STROKES PER MINUTE.

Diameter	STROKE.														
	2	3	4	5	6	7	8	9	10	12	14	16	18	20	24
1	0.81	1.21	1.63	2.02	2.42	2.85	3.26	3.68	4.10	4.90	5.70	6.52	7.36	8.20	9.80
2	3.26	4.89	6.52	8.15	9.78	11.41	13.04	14.67	16.30	19.56	22.82	26 08	29.34	32.60	39.12
2½	5.06	7.63	10.18	12.72	15.27	17.81	20.36	22.90	25.45	39.54	35.63	40.72	45.81	50.90	61.08
3	7.34	11.01	14.68	18.35	22.02	25.69	29.36	33.03	36.70	44.04	51.38	58.72	66.06	73.40	88.08
3½	9.95	15.00	20.00	25.0	30.00	35.00	40.00	45.	50.	60.	70	80	90	100	120
4	13.1	19.6	26.2	32.6	39.2	45.8	52.2	58.9	65.3	78.4	91.6	104.4	117.8	130.6	156.8
4½	16.6	24.8	33.0	41.3	49.5	57.8	66.1	74.4	82.5	99.0	115.6	132.2	148.8	165.0	193.0
5	22.0	30.6	40.8	50.9	61.1	71.3	81.5	91.6	101.8	122.2	142.6	163.0	183.2	203.6	244.4
6	29.4	44.0	58.8	73.4	88.0	103.	117.6	132.3	146.8	176.0	206.	235.0	264.6	203.6	352.0
7	39.9	60.	79.8	100.	120.	140	160	180	200	240	280	320	360	400	480
8	52.2	78.4	104.4	130.8	156.8	183.	208.8	234.8	261.6	313.6	366.	417.6	469.6	523.2	627.2
9	66.1	99.2	132.2	165.	198.4	237.	264.	297.	330.	396.8	474	528	594	660	793
10	81.5	122.4	163.	204.	244.8	286.	326.	367.	408.	489.	572	652	734	816	978
12	117.6	176	235.2	293.6	352.	412.	470.	529.	587.	704.	824	940	1058	1174	1408
14	160.	240	320.	400.	480.	560.	640.	720.	800.	960.	1120	1280	1440	1600	1920
16	209	313	418	523	626	732	836	939	1046	1252	1464	1672	1878	2092	2504
18	264	397	528	660	794	948	1056	1188	1320	1588	1896	2112	2376	2640	3176
20	326	489	652	816	978	1144	1304	1468	1632	1956	2288	2608	2936	3264	3912
22	395	591	790	988	1180	1380	1580	1780	1978	2360	2760	3160	3560	3956	2720
24	470	704	940	1174	1408	1648	1880	2116	2348	2816	3296	3760	4232	4696	4832



COMPOUND DUPLEX PUMP FOR TANK SERVICE.
The Deane Steam Pump Co.

and the speed at which it runs. Approximate values are shown in table VI.

TABLE VI.

Class of Pump.	Slip in Percent of Volume Pumped.	
	At high speed.	At low speed.
Small Centrifugal...............	75	60
Medium "	50	40
Fire Engines..................	40	30
Large Centrifugal...............	25	15
Lift and Drainage...............	15	10
Mine and Deep Well Pumps.....	8	5
Waterworks.....................	6	4

We can now determine the size of a pump to move a given volume of water. Suppose, for example, that it is required to force 200,000 pounds of water through a condenser per hour. This, at 8.338 pounds per gallon, would be 23,987 gallons per hour, or 400 per minute, if there were no slip; but for a common lift pump we must allow 10 per cent so that the capacity must be for 400×1.10=440 gallons per minute. From table V this would require a 10×12, a 9×14 or an 8×18; the 10×12 would be the most common proportion.

The area of valves should be such that the velocity of the water through them will not exceed 250 feet per minute; for 440 gallons, the volume would be $\frac{440 \times 231}{1728} = 58.8$ cubic feet. At 250 feet per minute, the valve area $= \frac{58.8}{250} = .2352$ square feet, or practically 34 square inches. The area of 3-inch valve is 7 square inches, so that the five valves would be sufficient. The circumference is 9.42 inches; this gives a lift of $\frac{34}{5 \times 9.4} = .723$ inch, an amount perhaps not too great for smooth action at the low pressure of a condenser pump; but it would be better even for this service to use twelve 2-inch valves giving a lift of .45 inch.

To find the power required, it is necessary to know the pressure against which the pump is to act both for suction and forcing. Assume this to be, in the present case, 5 feet of suction head and 5 pounds pressure due to the friction in piping and condenser tubing; the total pressure will then be 7.16 pounds per square inch, and on the 10-inch piston the total force would be 78.54×7.16 = 562 pounds. The pump is to make 120 strokes per minute, each stroke being 12 inches or one foot long, so that the work per minute would be 120×562 = 67,440 foot pounds, and the power 67,440÷33,000 = 2.04 horsepower provided there were no friction. It is customary to allow 25 per cent for friction in pumps, which would increase the power required to 2.04×1.25 = 2.55 horsepower.

To find the diameter of steam piston needed, the steam pressure must be known and the foregoing process can then be reversed. Assume the pressure to be 80 pounds; the work per minute is 2.55×33,000 = 84,000 foot pounds, which, at 120 feet per minute would require a force of 700 pounds, and, at 80 pounds per square inch an area of $\frac{700}{80} = 8.75$ square inches, corresponding to about a $3\frac{3}{8}$-inch diameter. Usually the steam and water ends are made much more nearly the same size than this would indicate and the steam throttled between the boiler and the steam end of the pump, so that the pump may be adaptable to a wide range of conditions.

The parts of a pump should be proportioned to withstand the forces which are to come on them, not only without danger of breaking, but without bending. For the parts which are to be of cast iron, such as cylinder, valve-chests and frame, the consideration which decides the dimensions is often the thickness of metal which will cast well; while for the piston it is the length which will give a good bearing surface on the cylinder to resist wear, and a tight joint to prevent leakage. These points are largely matters of experience and good judgment, but some ideas can be given as to common practice.

For a packed piston, usually 4 rings of $\frac{3}{4}$-inch square packing are used. Common values of rod diameters are given in Table IV. In designing the body and follower of the piston, it should be remembered that the less weight the better so long as the strength is sufficient; but a piece of cast iron less than $\frac{3}{8}$-inch thick seldom casts well in any complicated casting.

For the cylinder, the same equation may well be used for both steam and water ends. Whitham gives, allowing for strength, rigidity and possible reboring: $t = .03\sqrt{P\ D}$ where P is the maximum pressure in the cylinder in pounds per square inch and D the diameter in inches; t is of course the thickness of the cylinder walls. The length of the cylinder will be the length of stroke plus the thickness of piston and fastening nuts plus twice the clearance at one end which should be at least ½ inch. The clearance should be counterbored to leave the length of the bore somewhat less than the length of stroke plus the length of piston. The soft packing or piston rings should not, however, be allowed to run into the counterbore. An average of various formulas for the thickness of cylinder heads and flanges as given by Kent, is $t = .00036\ P\ D + .31$ inches. The flanges on the cylinder are made of the same thickness as those on the cylinder head.

For large cylinders it is usual to support the head by ribs running from the center to the circumference, though some designers consider this bad practice. The flange through which the bolts pass is made ⅓ thicker than the thickness of heads given by the above formula. To avoid springing of the flanges, cylinder-head bolts should be so spaced that the distance on centers will be not over 4 to 5 times the thickness of the flanges. The diameter should be such that the stress in the bolts will be less than 5000 pounds per square inch. If N is the number of bolts and d their diameter,

$$N \times \frac{\pi d^2}{4} \times 5000 = \frac{\pi D^2}{4} P, \text{ and simplifying,}$$

$$d = D\sqrt{\frac{P}{5000\ N}}$$

The area of inlet and discharge ports and pipes should be at least equal to that of the valves.

The thickness of the walls of the valve chambers is usually a little greater than that of the cylinder walls. For high-pressure pumps it is best to give these chambers as nearly as possible a cylindrical form in order to secure the greatest strength for a given weight of material.

For the frame, it is impossible to give any rules or suggestions except that the student should get catalogues from the prominent makers and study them to see what has been found satisfactory.

For proportioning the driving end of a pump, whether steam, belt or motor driven, the ordinary rules for the kind of machinery involved apply. However, it should be remembered that the parts are subjected to severe shock on the reversal of the pump, hence a factor of safety as high as 15 should be used.

Where pumps are required for special service, the amount of water to be pumped per hour is generally fixed by the conditions of the service; but for a few cases the method of determining that amount can be definitely given. For a **boiler-feed pump,** it is necessary to know the horse-power of the boilers to be cared for by the pump. One boiler horse-power will call for the handling of 30 pounds of water per hour, and the pump should be specified accordingly with an allowance for good measure. For an air pump to remove the air and condensed steam the amount of fluid to be handled is difficult to estimate, hence, a common rule is to make the volume of the double-acting air-pump cylinder $\frac{1}{20}$ that of the low-pressure cylinder of the engine which it is to serve. The double-acting circulating pump has a volume about $\frac{1}{35}$ that of the low-pressure cylinder, or it may be figured from the horse-power of the engine, assuming the steam consumption at 25 pounds per I. H. P. per hour, and the cooling water at 30 pounds per pound of steam. Of course for accurate calculation, the steam consumption would vary with the size and type of engine, but the above will give safe average values.

Fire pumps are designed with special care to meet the requirements of the insurance companies. Stated briefly these are: Ability to start instantly after long disuse; must be duplex, with cross-operated valves; should preferably be of plunger pattern; must be brass-fitted throughout; designed to carry a water pressure of 320 pounds; suction valve area should be not less than 50 per cent of plunger area for 10-inch stroke and 56 per cent for 12-inch; discharge valve area should not be less than $\frac{2}{3}$ the suction-valve area; valve springs should be cylindrical, not conical, and held at the ends in a groove; studs to be so designed as to always allow a lift $\frac{1}{5}$ of the valve diameter; valve seats should be of gun metal held in the deck by a taper thread, or a smooth taper bore forced in, having the lower edge turned over; least area of exhaust passage should be 4 per cent of piston area; admission ports should be not less than 2½ per cent of piston area; clearances should be as small as possible; valve tappets should be non-adjustable; cushion valves controlling a by-pass from steam to exhaust port to regulate the amount of steam cushion at ends of stroke are recommended for 750 and 1000 gallon pumps; a gauge should be provided which will show at all times the length of stroke that the pump is making; water-pressure gauge with ¼-inch lever cock should be connected close to the air chamber; a relief valve of Ashton, Crosby or similar pattern and of size sufficient to discharge the full throw of the pump working at ⅔ speed should be furnished and set to 100 pounds pressure; this valve should have a hand wheel conspicuously marked showing the direction to ← OPEN ; relief valve should discharge downward into a vertical pipe, thence into a funnel; ⅜-inch brass drip cocks with lever handles should be on both ends of the steam and water cylinders; a ¼-inch lever air-cock should be on the cover over the water cylinders; each pump should be fitted with brass priming pipe, starting with a 2×1×1 inch brass tee close to the pump beneath the delivery flange and leading to four ¾-inch valves, one opening to each of the four plunger chambers; a priming tank should be provided with bottom at least 5 feet above the pump, and having a capacity half that of the pump in gallons per minute, unless water flows to the pump under pressure; priming tank must be used only for that purpose; each pump is to be fitted for hose connections as per number of streams to be served; capacity of cylinders, sizes of piping, air and suction chambers and relief valves should be as per Table VII.

A special acceptance test is demanded as follows: The pump must run smoothly and without slamming at its full rated speed, maintaining a water pressure of 100 pounds per square inch when furnished with steam at a pressure of 45 pounds for the 500-gallon pump, 50 pounds for the 750-gallon pump, 55 pounds for the 1000-gallon pump. The water to be discharged through 1⅛-inch nozzles on hose lines 150 feet long; the hose must lie quiet, showing uniform delivery; with all water outlets closed and steam supplied to give 80 pounds water pressure, leakage must not allow

TABLE VII.

Dimensions and Capacities of Fire Pumps.

No. of $1\frac{1}{8}$-inch Streams.	Capacity, Gals. per min.	Ratio of Piston areas.	Boiler H. P. needed.	Steam press. needed at pump. Lbs. per sq. in.	Nearest commercial Size. Inches.		
					Diam. steam cylinder.	Diam. water cylinder.	Stroke.
One. (Too small except for auxiliary.)	320	4 to 1	....	40	10 12 10 12	5 6 5 6	10 10 12 12
Two. (For small mills.)	500	4 to 1	100	40	16 14	8 7	10 12
Three. (For general use.)	750	3 to 1	115	48	16 16	$9\frac{1}{4}$ 9	10 12
Four. (For large factories.)	1,000	3 to 1	150	44	$18\frac{1}{2}$ 18	$10\frac{1}{2}$ 10	10 12

For over 1,000 gallons per minute, two pumps should be used.

Capacity, Gals. per minute.	Diameter of pipes in inches.				Capacity, gallons.		Diameter Relief Valve, Inches
	Suction.	Discharge.	Steam.	Exhaust.	Air chamber.	Suct. cham.	
320	6	5	$2\frac{1}{2}$	3	10	8	$2\frac{1}{2}$
500	8	6	3	4	17	13	3
750	10	7	$3\frac{1}{2}$	4	25	18	$3\frac{1}{2}$
1,000	12	8	4	5	30	24	4

the pump to make more than one double stroke per minute; with water outlets nearly closed and pump running slowly it must carry water pressure of 240 pounds per square inch and all joints remain practically tight; with relief valve set at 100 pounds and all other outlets closed it must discharge full delivery at 50 double strokes per minute with the water pressure rising to not over 125 pounds.

These specifications call for a first-class design and construction, but such are necessary if a fire pump is to serve its purpose of a protection in case of emergency.

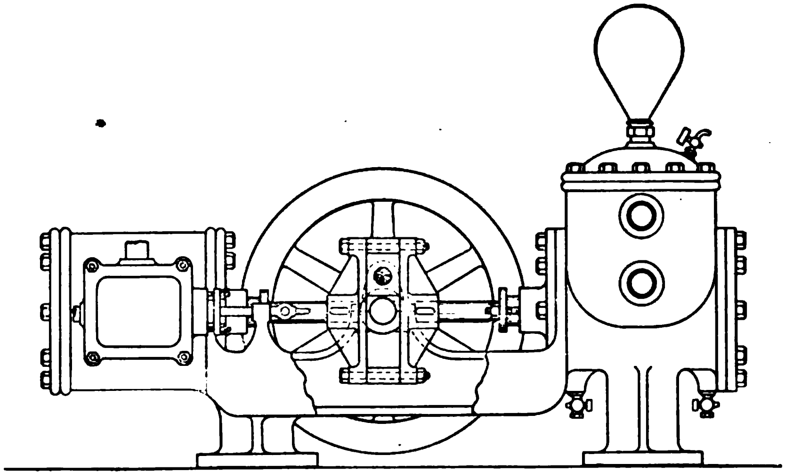


Fig. 36.

A name plate must be placed on the inboard side of the air chamber bearing data as follows in black enamel letters ½ inch high on a white enamel ground:

Diameter of cylinders and stroke 16×9×12. Capacity gallons per minute 750 or three 1⅛-inch smooth nozzle streams. Full speed, revolutions or double strokes. "For Fire purposes never let steam get below 50 pounds, nights or Sundays."

TYPES.

In classifying pumps by types, two methods of division naturally suggest themselves; the first according to the arrangement of the water end, the second according to the means of driving employed. The difference between single-acting and double-acting pumps has already been noted; the single, duplex or triplex arrangement, depends on whether one, two or three cylinders are assembled into a single machine, drawing water from the same suction inlet and discharging it into the same outlet.

The single pump is much used for small sizes or where constancy of flow is of minor importance. It is, of course, the cheapest arrangement and gives satisfactory results if the pump is of ample size for the work. A difficulty with this type of pump is the provision of a means of operating the steam valve to reverse the motion of the piston at the end of the stroke. The pistons in steam and water cylinders must be stopped gradually to avoid pounding, and this leaves no force available just at the end of the stroke when it is needed to move the steam valve so as to reverse the motion.

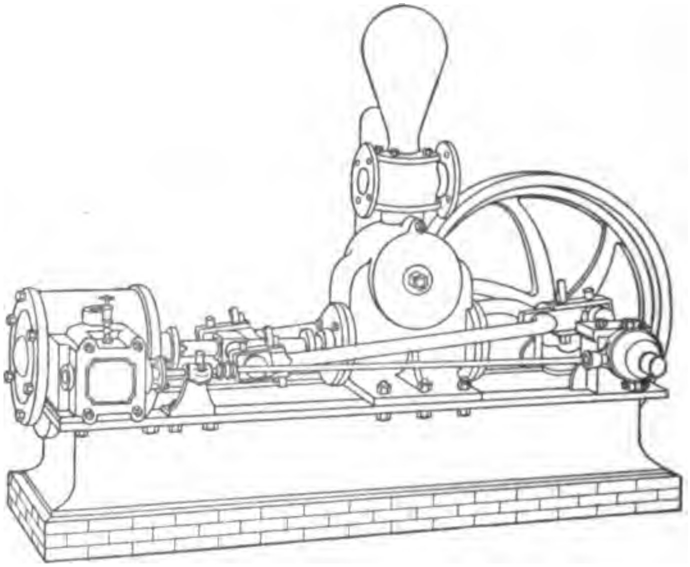


Fig. 37.

The difficulty has been overcome in two ways; first by the introduction of a shaft and fly wheel, driven by a crank and a yoke in the rod between the steam and water cylinders as seen in Fig. 36. The inertia of the fly wheel then furnishes the force needed to move the valve at the instant of reversal, generally by means of an eccentric on the shaft. This form is often used for fire-engines, two or three sets of cylinders sometimes being connected to one shaft, in which case the pump becomes of duplex or triplex form. For stationary pumps, the fly wheel and shaft are more commonly placed beyond the cylinders at one end of the frame, and the shaft is driven by a crank and connecting-rod mechanism from a crosshead attached to the piston rod as seen in Fig. 37. The fly wheel also makes it possible to use steam expansively. The second method of effecting reversal is by the use of an auxiliary valve and steam-driven main valve. This is the system generally found on small, single-cylinder, direct-acting pumps for boiler feeding, tank filling and similar uses. The details of the devices used vary greatly with different makes; these will be taken up later.

The single pump is compact, cheap and convenient, but the flow of water is by a series of impulses rather than in a steady stream, so that, if the pump is forced, it slams badly and produces water hammer, an evil but partially remedied by the use of discharge and suction air chambers.

Duplex pump. To avoid this, and at the same time to simplify the moving of the steam valve, the late Henry R. Worthington devised the duplex form, in which two direct-acting pumps are mounted side by side, the water ends and the steam ends working in parallel between inlet and exhaust pipes as seen in Fig. 38. The steam valve for cylinder A is moved by a bell-crank lever driven from the rod of cylinder B as it nears the end of its stroke; so that, when the piston of B is about to come to rest, that of A is set in motion. While the piston of A makes its stroke, that of B is at rest until that of A, near the end of its stroke, moves the steam valve of cylinder B by means of a second bell-crank lever, and the piston of B is once more set in motion. Thus the pistons move alternately, but one or the other is always in motion and the flow of water is made practically continuous. By this arrangement the auxiliary steam valve is made unnecessary and a simple D valve, driven by a valve rod running through a stuffing box, is used.

The triplex pump having three cylinders delivering to the same outlet is generally used for power pumps; the pistons or plungers are driven from a shaft by three cranks or eccentrics, set at 120 degrees to each other so that the turning effort required on the shaft may be as uniform as possible and the flow of water steady. A style often used is single-acting with trunk plungers, that is, the connecting rod which drives the plunger is fastened to a pin inside the hollow body of the plunger as seen in Fig. 39. This gives a compact form and one not likely to get out of order, but the trunk arrangement is inconvenient in case of any wear on the pin inside the plunger, and the side thrust of the connecting rod creates a certain tendency to wear the plunger packing so that it will not remain tight. On the whole it is better, though more expensive, to have a regular crosshead and ways to receive the thrust of the connecting-rod as in Fig. 42. The cylinders in this style of pump are usually vertical and the crank shaft is driven through a pair of reducing gears from a second shaft which is, in turn, driven by a pulley and belt or an electric motor.

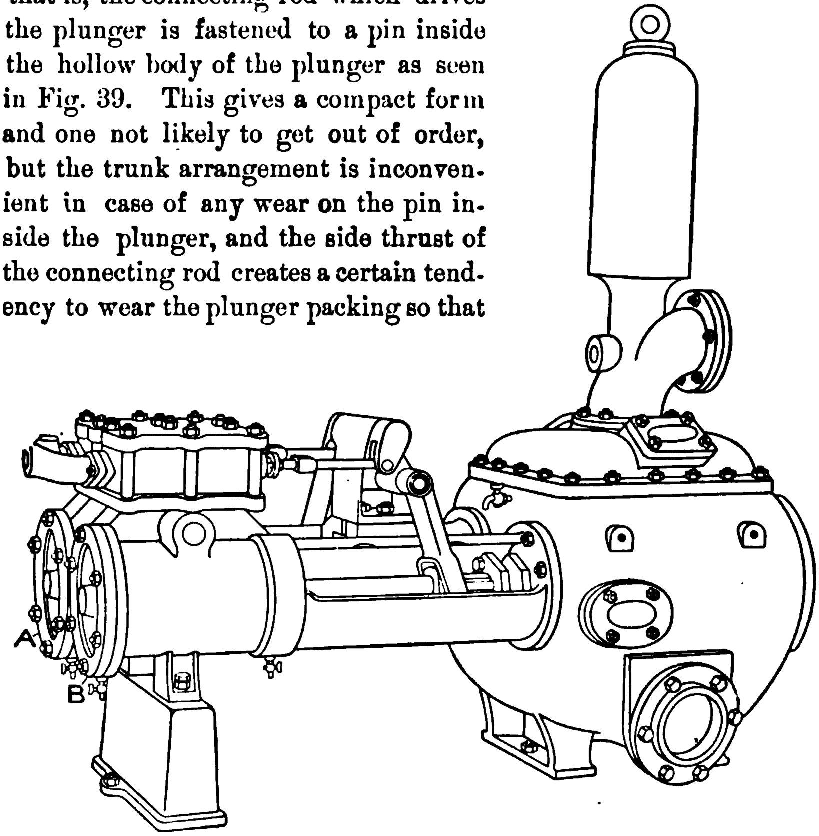


Fig. 38.

Methods of Driving. With respect to the method of driving, the most common type of pump is the direct-acting, steam-driven, with steam piston and water piston or plunger on opposite ends of the same rod. For small sizes, the steam cylinder works without expansion of steam, the piston being driven by full boiler pressure throughout the stroke. In fact it is impossible to avoid this feature of full-stroke admission, since the pressure in the water end is constant throughout the stroke, unless some means, such as a fly wheel or accumulator, be provided, a complication which is undesirable with small powers. The simplest way to make any use of the expansive force of the steam is by compounding; that is, letting the steam exhaust from the small, high-pressure cylinder, into which it is first admitted, to a larger low-pressure cylinder, placed in tandem, with the first one. See Fig. 40. This arrangement gives a pressure on the low-pressure piston, which gradually decreases from the beginning to the end of the stroke while the pressure on the high-pressure piston is constant, so that whatever useful work is gained from expansion is in the low-pressure cylinder. This is, however, found to be sufficient to pay for the extra cost if the service is continuous, in sizes having a water cylinder over 8 × 12 inches. The compound pump is nearly always duplex, though a few manufacturers offer a single tandem-compound style. Since the steam pistons must move together, the steam valves must move together, and a single motion serves to operate the valves for both high and low-pressure cylinders.

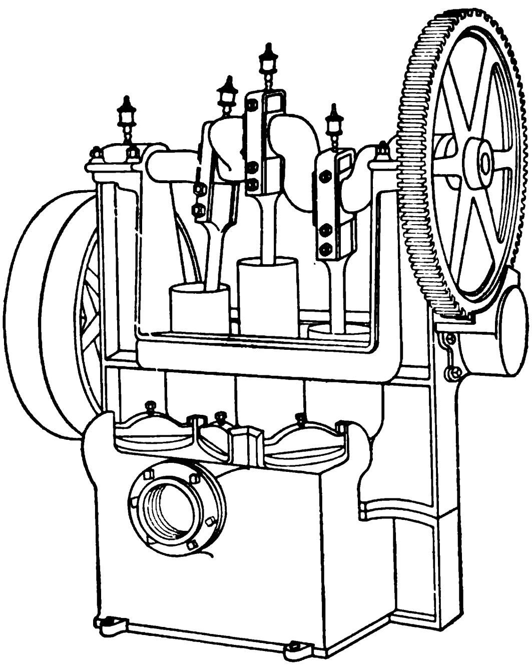


Fig. 39.

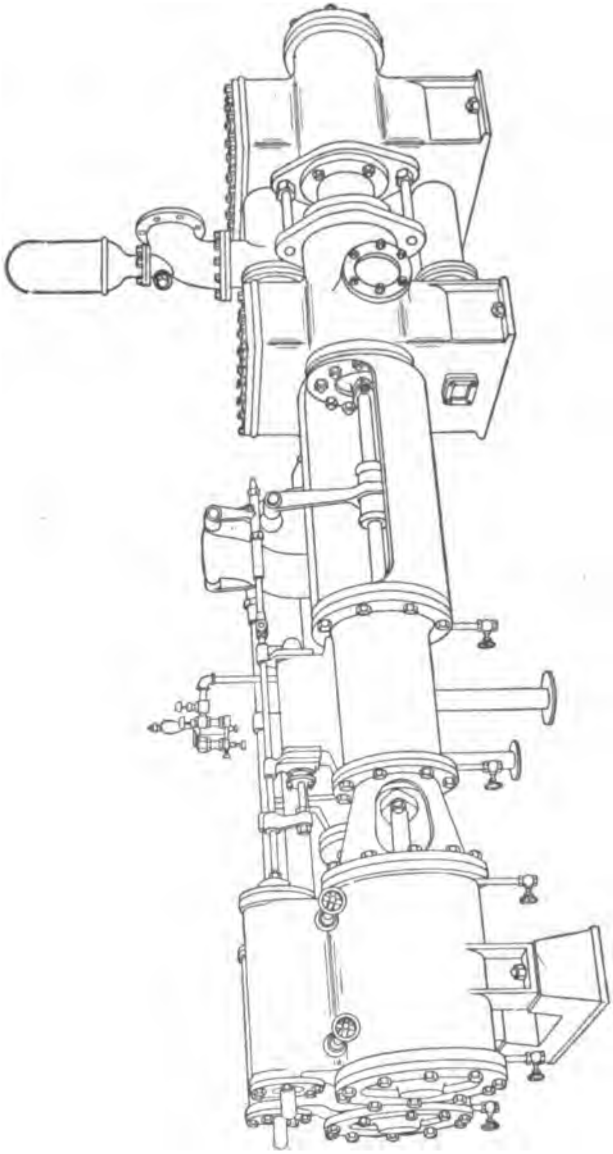


Fig. 40.

The simple steam-driven pump requires from 60 to 120 pounds of steam per horse-power, per hour; the gain by compounding is about 30 per cent per 100 pounds increase in boiler pressure, so that the steam consumption will be 40 to 95 pounds per horse-power hour.

The volume of the low-pressure cylinder is made from 2 to 4 times that of the high-pressure, depending on the initial pressure to be used.

The large steam consumption of direct-acting pumps of the single-cylinder type has led to the adoption, in many large stations, of power-driven pumps of various kinds. It is probably true that the cost for steam is less by this method when the power is developed by large engines, even after the losses in transmission are accounted for, because the efficiency of the engine is so much greater than that of a small pump. But, for small plants, the convenience of having the pump located near its work and of being able readily to control the speed has overbalanced the consideration of saving.

As previously mentioned, nearly all power pumps are geared, the power being applied to a driving shaft as seen in Fig. 39. The belted type is driven from a countershaft and provided with fast and loose pulleys used for starting and stopping, but there is usually no means of speed regulation. This, together with the expense of belting, the room needed and the fact that the convenient location for a pump is likely to be such as to make a belt a nuisance have prevented a general use of this type, and with the general introduction of electric power into all plants, the motor-driven pump is superseding it.

The motor-driven pump is of the same general form as the belted, but has an electric motor mounted on the driving shaft in place of the pulley, or has a double-reduction gearing as in Fig. 41. The high speed at which motors run make this double reduction necessary except in the case of large pumps. The only alternatives are the use of a large, slow-speed motor or a belted motor, either of which would be cumbersome and expensive. As to efficiency, probably there is not much difference between the belted and the motor-driven types. The electric method of transmitting power from the engine to the pump will be more economical than belting, but the transformation in the motor and the loss in the extra set of gears will offset this.

The speed of the motor can, however, be controlled by a starting box, or rheostat in the field circuit, hence the motor-driven has a considerable advantage over the belted type.

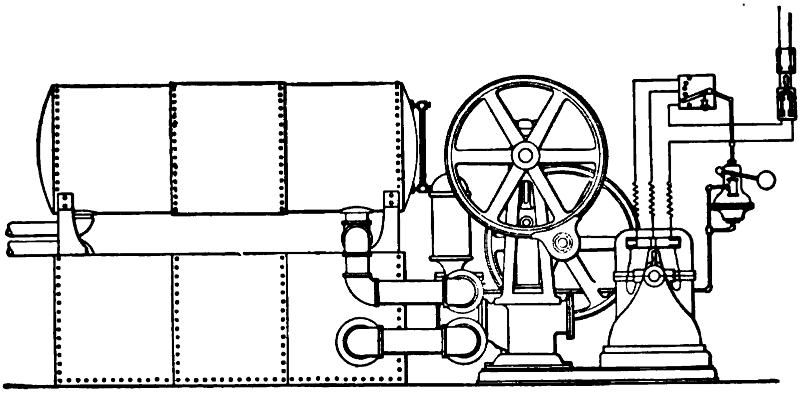


Fig. 41.

As compared with the direct-acting type the economy would be about like this:

Direct-acting—steam consumption, average, 90 pounds per hour per H. P.

Belt-driven—steam consumption of engine, per B. H. P. hour, average 35 pounds; at an efficiency of transmission for belting of 70 per cent and for gearing of 85 per cent, this would require about 59 pounds of steam per hour per H. P. delivered to the pump, and at an efficiency of 75 per cent for the pump this would give about 78 pounds of steam per hour per H. P. of effective work.

Motor-driven—steam consumption as before, 35 pounds; efficiency of dynamo 93 per cent, of transmission 90 per cent, of motor 93 per cent, of gearing 72 per cent, of pump 75 per cent, gives a steam consumption per effective H. P. of 83 pounds per hour.

Of course with large units the efficiencies may be better, but the relation would be about the same. The greater convenience of the direct-acting pump as to size and means of control, and the ability to start it when the engine is not running have weighed in its favor in the majority of cases.

For some few places a gas or oil engine has been used for pump driving. In this type a clutch must be introduced between the engine and the pump, Fig. 42, as the engine will not start under load. The engine must first be started, then the pump thrown on empty and finally the valves operated so that the pump will begin to work. There is no means of varying the speed, but a regulator is sometimes introduced which throws out the clutch when the water reaches a certain height in the tank or reservoir to be filled. This type is used only for such service as water supply for a house or reservoir system, and would hardly be desirable elsewhere.

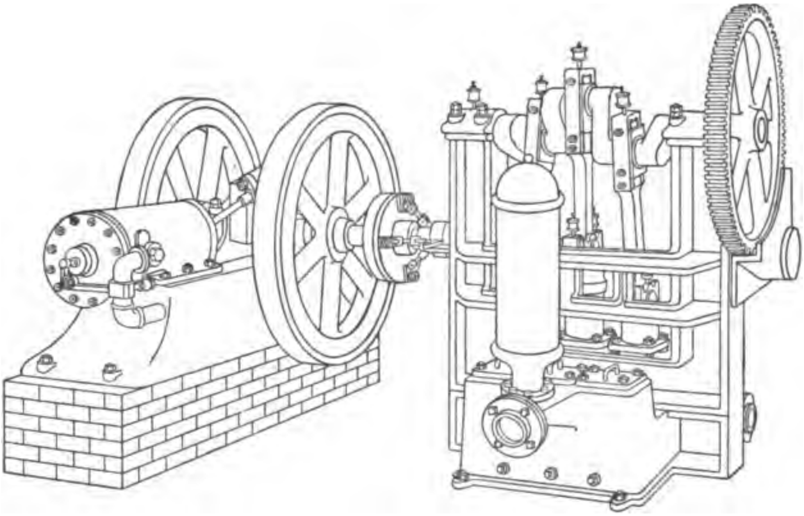


Fig. 42.

To economize in floor space, the vertical marine type of pump is used. This is generally duplex, but occasionally single, and except for the vertical arrangement is identical with the horizontal type.

Another special type is the **combined-condenser pump**, Fig. 43, in which the steam cylinder is placed in the center and drives on one end of the piston rod a vacuum pump for drawing the condensed steam and air from the condenser, and on the other end a pump for forcing the cooling water through the condenser tubes. The steam cylinder is the same as for any single pump except that there is a stuffing box at each end; the water end is the same as for any pump; but the air-pump end has composition disc valves with an area of opening larger than that for a water cylinder of the same size, and the springs which close the valves are set at a less tension than for water.

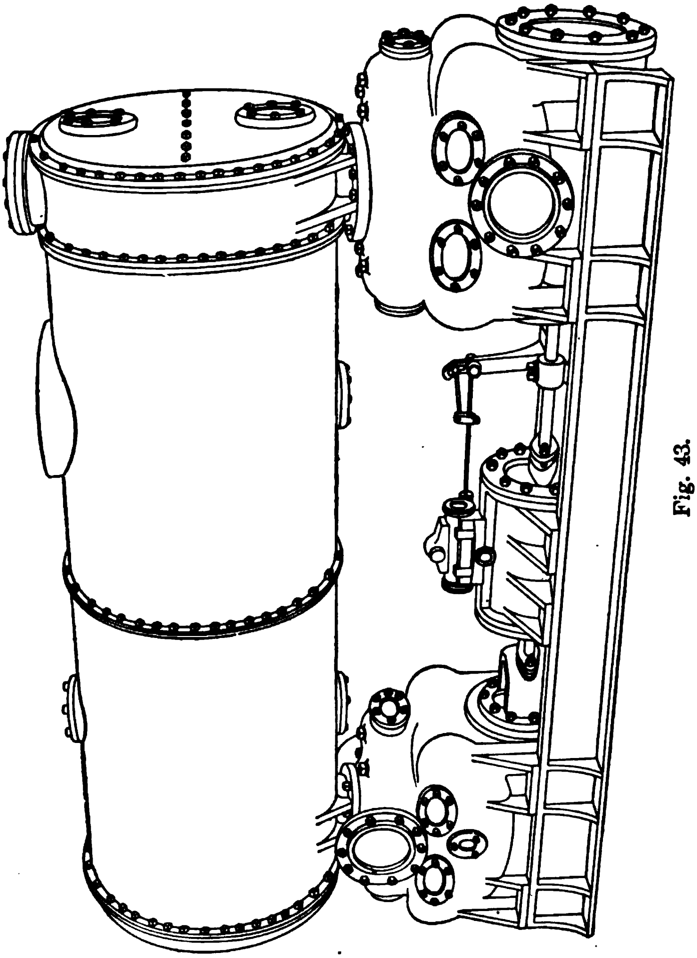
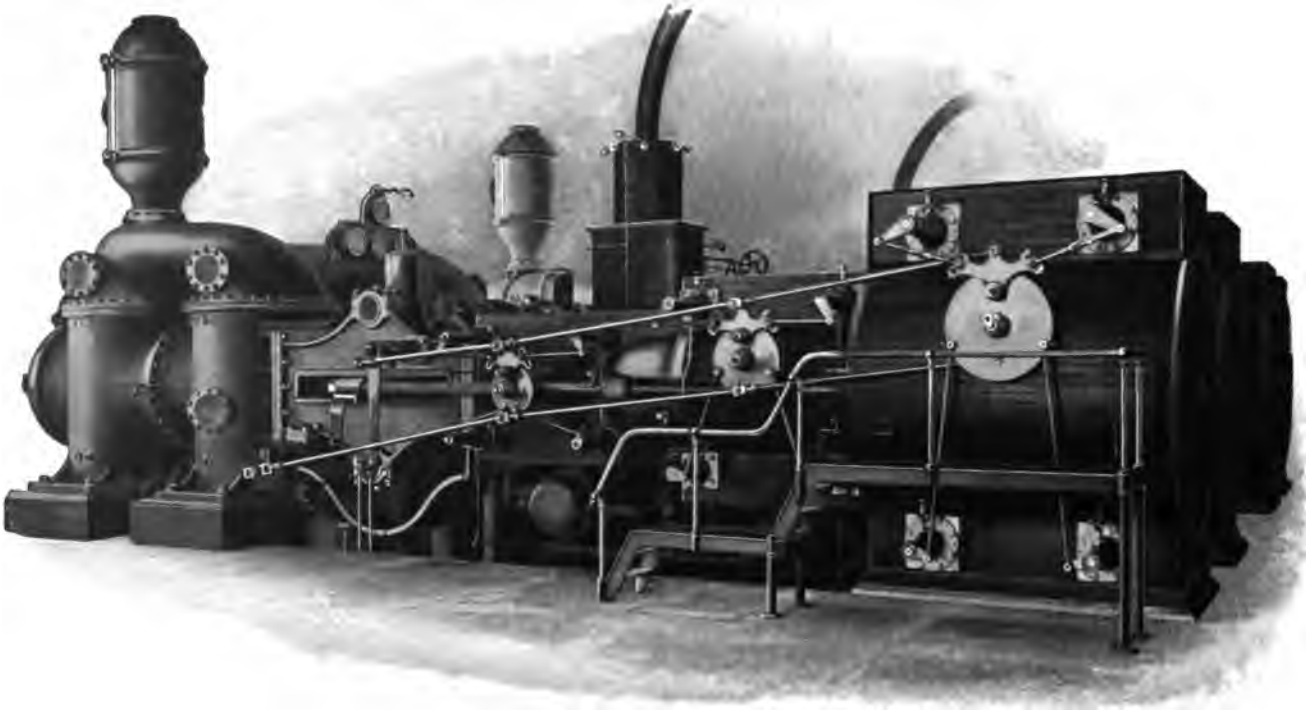


Fig. 43.

Air pumps of special forms for condensers are often used combined with the condensers; they are driven by a separate engine or by a bell-crank lever from the crosshead of the main



WORTHINGTON HIGH DUTY TRIPLE EXPANSION PUMPING ENGINES.
Mt. Royal Pumping Station, Baltimore, Md.

engine. The cylinders are often vertical and single-acting on the lifting principle with valves in the top of the bucket. A unique and in some ways commendable style is that of Fig. 44 which avoids inlet valves and valves in the bucket, yet removes both air and the water of condensation quickly and effectively. Condensed steam and air flow into the lower recess A through the passage B from the condenser; as the piston descends, it forces the water from A so that it shoots up in the direction indicated by the dotted lines through the ports *p p* into the cylinder C. The piston rising, closes ports *p p* and catches the water and air in the cylinder forcing them out through the valves *v* in the top into the discharge chamber D, whence the mixture escapes through the exhaust passage K to the hot well, or wherever may be desired.

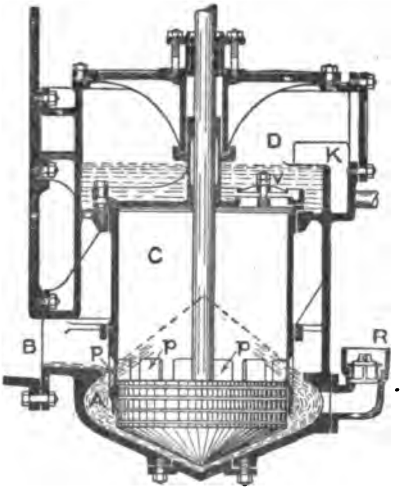


Fig. 44.

For certain uses, such as keeping up the pressure on a hydraulic elevator system, draining a pit or maintaining the water level in a tank, it is essential to have an automatic method of control which shall start and stop the pump as may be needed. For direct-driven steam pumps this is accomplished by control of a damper valve or a quick-acting gate placed in the steam pipe and for the motor-driven pump by a switch on the starting resistance box. In the one case, a float is moved with the water level, in the other a diaphragm worked by the tank pressures actuates a series of levers as seen in Fig. 41 so as to work the steam valve or switch as the case may be.

For high pressures and high speeds what is known as the Riedler System of mechanically operated valves (Fig. 45) is found especially desirable. The valve is of the poppet type, made so large that one valve answers for each end of a cylinder and is opened and closed by a rocking arm N, turned by a link and driven from a wrist-plate much as in a Corliss-engine valve motion. The mechanical driving permits a high lift, with rapid opening and closing at the end of the stroke, thus decreasing the slip and shock. The high lift, 1 to 2 inches, and large water passages decrease the friction while the mechanical operation of the valves also allows of the high piston speed necessary to secure economy in steam consumption.

Referring to Fig. 45, the valve body P has on its lower face a leather seal S secured by a plate O. The sleeve D works on the spindle B, raised by a pin in the arms K which passes under the collar H and is lowered by the pressure of the blocks J on the cushion plate G. F is a rubber buffer to take the shock in case an obstruction gets into the valve. The arms K are mounted on and turned by a spindle which passes through the bushing C and is rotated by the crank L and crank pin N. One valve is used for inlet and one for outlet on each cylinder, as seen in the right-hand plunger of Fig. 46; for the left-hand end, the outlet valve of the larger plunger acts as an inlet and the clack valve between passages

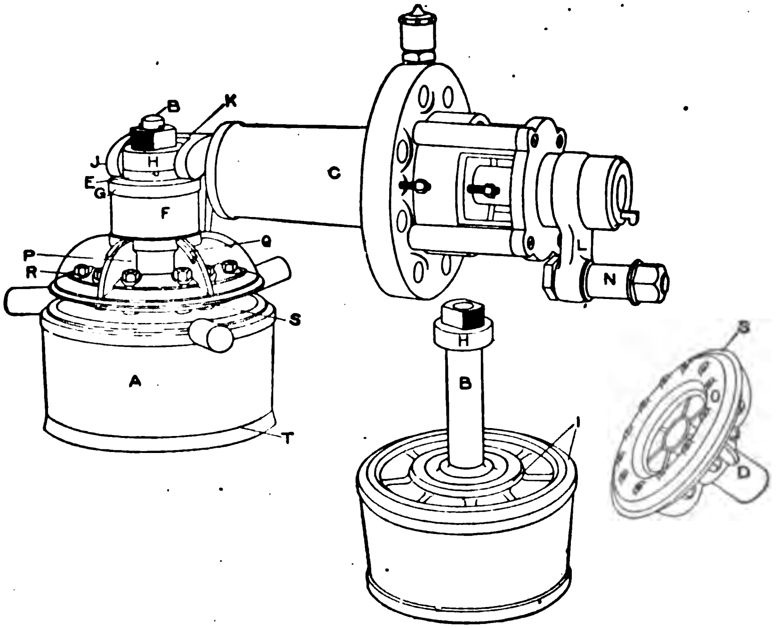


Fig. 45.

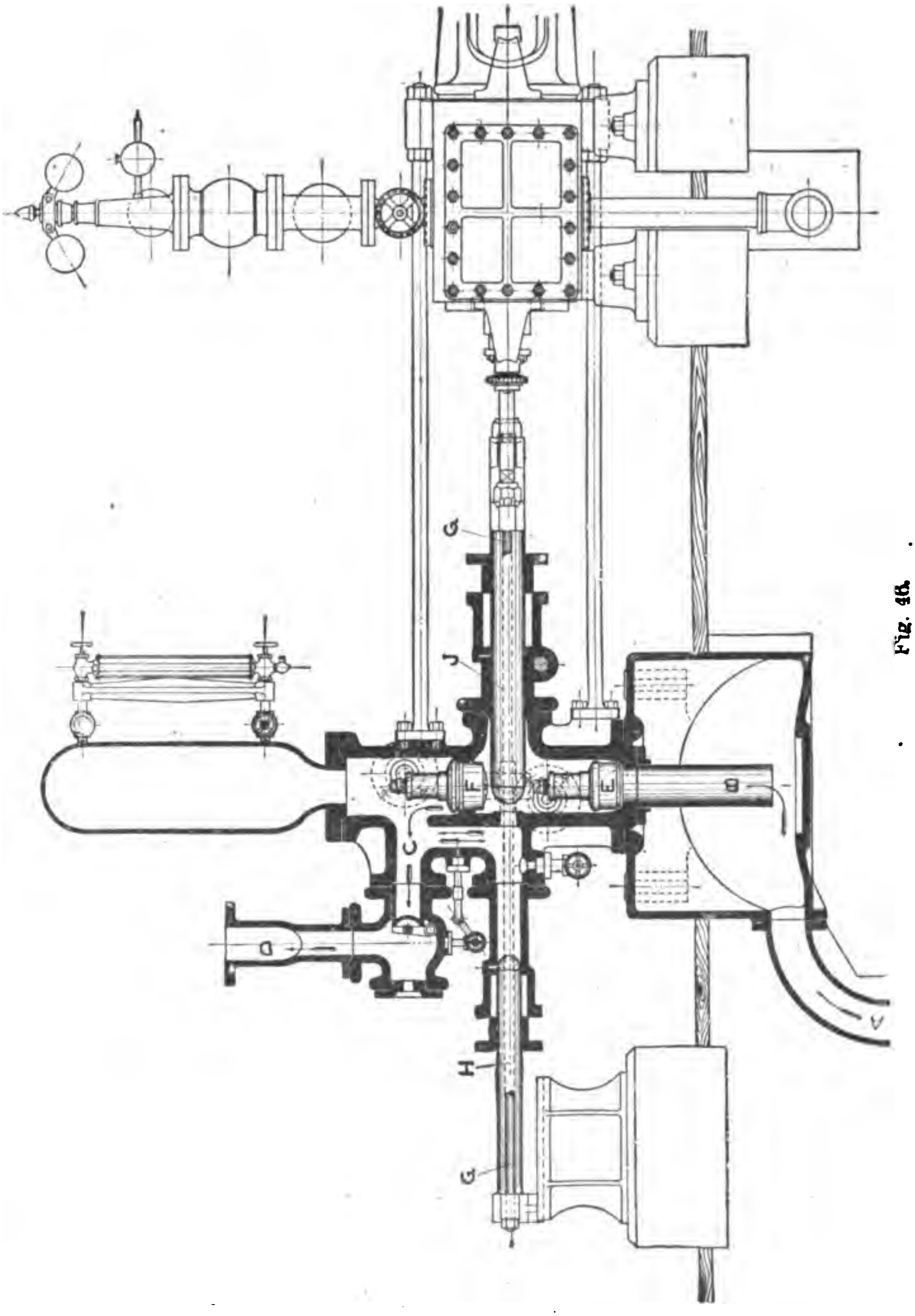


Fig. 46.

C and D acts as an outlet valve. The differential type of pump here shown is used for high pressures and to give a continuous flow with few valves; the cross-section of plunger H (Fig. 46), is half that of J, hence, on the stroke towards the left, half the water discharged by J is thrown into the small plunger chamber and half into the passage D; on the stroke towards the right, the half which entered the small plunger chamber is driven into chamber D and the large plunger draws its full quantity through the suction pipe B. The large passages and valves, the ample discharge and suction air-chambers, together with the positive opening of the valves, combine to make this pump very efficient and quiet in running. However, it requires a large space in proportion to its capacity while its complexity makes it expensive, so that it is available only for large sizes and in places where the space occupied is not of special importance.

STEAM VALVES.

The simplest form of valve is that used with a fly-wheel pump and which is moved by an eccentric as seen in Fig. 37; the detail is shown in Fig. 47 and the action as follows:

With the valve *v* in the position shown, steam is flowing through the port *a'* into the right-hand end of the cylinder and forcing the piston towards the left; at the same time, any steam which may be in the left-hand end of the cylinder is driven out through the port *a* into the exhaust passage *b* and thence to the exhaust pipe. If the shaft M be turning as indicated by the arrow, the eccentric *e* which is keyed to it, will also turn, and sliding in the eccentric strap S will draw it towards the right, and with it the valve. When the eccentric has made a fourth of a revolution it will be vertically above the shaft, and will have drawn the valve over so that it covers both ports, while, at the same time the piston will have moved so that it will stand at its extreme left-hand position. As the motion continues, the left-hand port *a*, will be opened to steam and the port *a'* to exhaust, causing the piston to reverse and move towards the right, which will continue until the eccentric has moved the valve to its extreme right-hand position and back to cover both ports. The eccentric standing directly below the shaft, the piston will again reverse and start towards the left, and the cycle be repeated.

This style of valve is called the D valve, and is much used for pumps and for the smaller reciprocating steam engines. For the ordinary pump the valve is made so that when in mid position it just covers the ports, and steam will enter one or the other end of the cylinder, if the valve is moved from that position. Steam will then be admitted full stroke and work without expansion in the cylinder. If it is desired to utilize some part of the expansive force of the steam, "lap" must be added to the valve as shown at *a*, Fig. 48, so that the port may be closed before the piston reaches the end of the stroke; then the eccentric must be turned forward on the shaft in the direction of its motion in order that the valve may open by the time the piston is ready to reverse, and then "inside lap," *b*, must be adjusted so that the exhaust port will neither open too soon, thus wasting the expansive energy of the steam, nor close too early, thus causing too great a compression of the steam caught between the piston and the end of the cylinder.

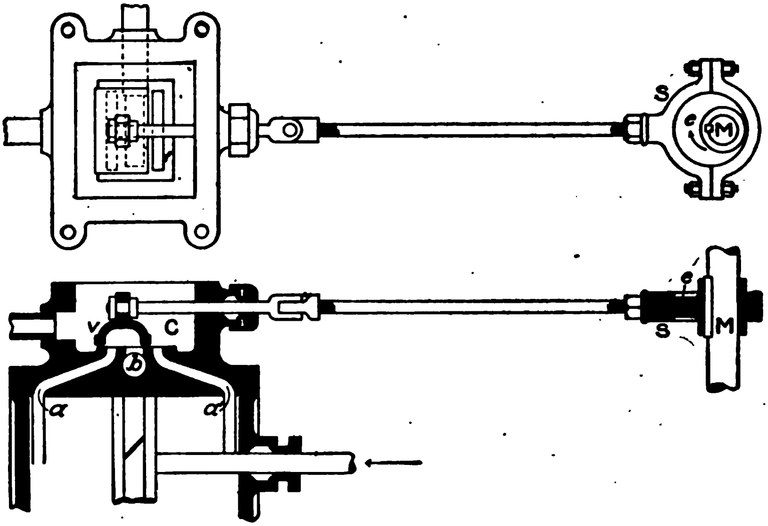


Fig. 47.

The construction is the same as for a simple reciprocating engine, which indeed it is, with the addition of a pump cylinder and piston attached to the end of the piston rod.

Another form of valve common in pumps is the B type, Fig. 49. The action is the same as for the D type, except that steam flows into the cylinder through the cup-shaped cavity in the valve instead of past the end, and the motion of the valve relative to that of the piston is the opposite of that for the D valve. The B type is adapted for use with the steam-driven valve to be described, but is not suited to be driven by an eccentric. The steam-driven valve is used on the great majority of small single pumps on account of the compactness, light weight and low cost and in spite of the poor steam economy. In a few of the first named pumps, for instance, the early Worthington, single type, the main steam valve was moved by tappets on the valve rod which were struck by an arm fastened to the piston rod. This was not certain in its working and has been displaced by the main valve moved by a small piston

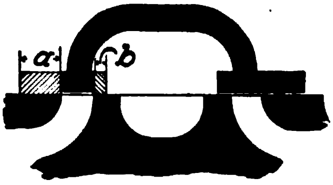


Fig. 48.

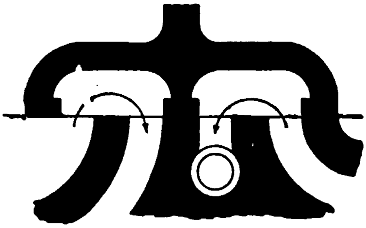


Fig. 49.

working in an auxiliary cylinder. Steam is admitted to this cylinder by an auxiliary valve moved by a tappet or lever mechanism worked from the main piston rod.

The designs of different makers vary more in this detail than in any other, and we shall now discuss a few typical examples at some length.

The first to be developed was the Knowles, Fig. 50. The chest piston A is moved by steam admitted through ports *m* and *z* between the end of the chest piston and its cylinder, and carries with it the main steam valve B, which opens and closes the ports to the main steam cylinder. The ports *x y z* and *x' y' z'*, of the small detail view, are made to register with the ports *m n* and *m' n'* by the turning of the chest piston by the tappet C driven by the rocker-bar R and the roller E. As the main piston reaches its left-hand position, the roll E will lift the left-hand end of the rocker bar and place the ports so that steam is admitted to the right of the chest piston and exhausted from the left; the main valve will then be carried to the position shown in Fig. 50 and steam be admitted to the left-hand end of the main steam cylinder in order to drive the main piston towards the right. At the other end of the stroke the operation will be reversed. The arm F will not strike tappets C or D unless the steam fails to move the main valve, in which case the piston A would be driven by pressure on the tappets, but with more or less slamming depending on the speed.

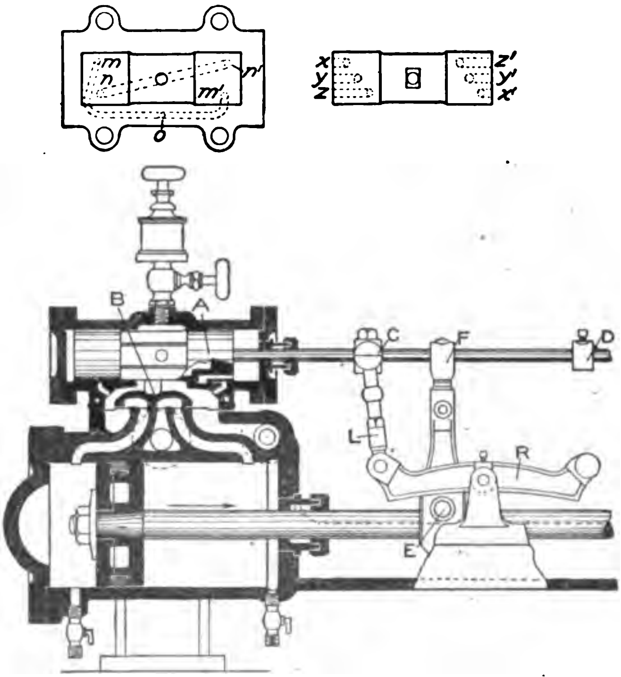


Fig. 50.

The third set of ports in the chest piston and its seat are to form a connection between the two ends of the auxiliary cylinder after the chest piston has moved a certain distance in order to prevent its striking the end of the chest. The length of stroke can be changed by altering the height of the roller E, and unevenness of stroke at the ends can be remedied by changing the length of the link L.

An entirely different style of valve is that of the Cameron pump, Fig. 51. F is the valve-piston which drives the valve G, by means of a projection which sticks up through the neck of the piston. The ends of the valve piston are hollow as shown in the cross-section of the right-hand end, and a small hole P, at each end admits steam at full pressure to the ends of the chest cylinder. If the main piston C is moving to the left (as it would be with the valve G in the position shown), when it reaches the end of the stroke, it will strike the reversing valve I′ and force it open, thus allowing the steam at the left of the valve piston to flow into the exhaust port through the passage E′. The steam at the right-hand end of the valve piston will expand, forcing the piston F to

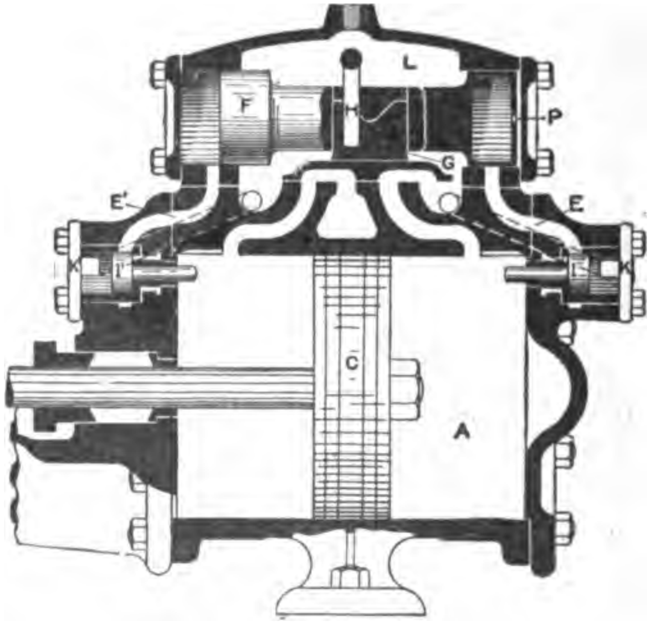
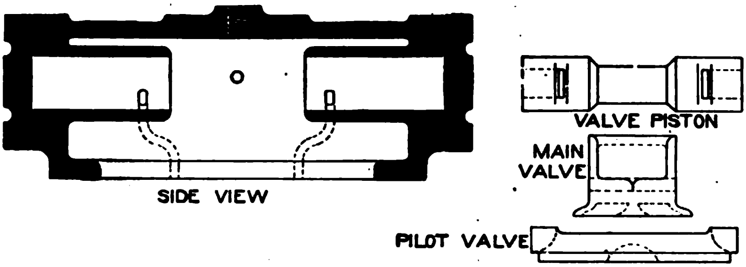


Fig. 51.

the left and carrying valve G with it, thus opening the right-hand end of the main cylinder A to the exhaust and the left-hand end to live steam, which will reverse the motion of the piston C. As the valve piston moves to the left, it will cover and close the port E′ hence will be brought to rest without shock by the cushioning of the enclosed steam. As soon as the piston C leaves it, the reversing valve I′ will be closed by the pressure in the passage K′ which is filled with live steam. At the other end of the stroke, the same series of operations follows by the action of the reversing valve I.

The valves require no adjustment and are easily brought to a new seat when they become worn, by removing the bonnets. If piston F becomes worn after long service, the cylinder may be re-bored and bushed or the holes P may be drilled a little larger, which will give greater pressure in the end of the chest cylinder and keep the pump working steadily for a long time even with the piston F considerably worn. H is a starting lever, worked by a handle outside, for use in case the pump should stop with the valve G covering both admission ports.



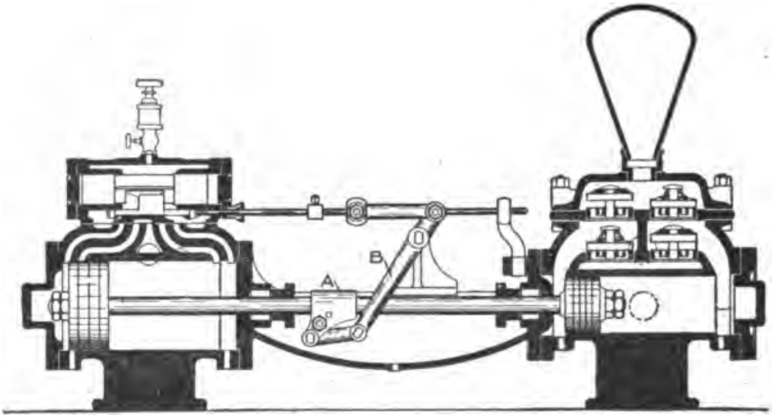


Fig. 52.

A third style of valve is shown in Fig. 52, that of the Deane of Holyoke. In this the auxiliary valve is in the form of a yoke surrounding the main valve; the working-valve seat and ports for the auxiliary are at one side of the main-valve seat, and the ports supply steam to or exhaust it from the ends of the valve-piston chamber through openings in the end of the piston. The auxiliary valve is moved by a valve rod actuated by tappets and a series of levers from the main piston rod. If the valve piston fails to move the main valve, the auxiliary valve, acting as a yoke, will finally drive the main valve over, by positive action.

A somewhat similar device is used on the Dean Bros'. pump, Fig. 53. There is, however, no period of rest of the auxiliary valve, the motion being similar to that given by an eccentric, and there is no provision for moving the main valve other than by the valve piston, because the positive driving of the auxiliary valve makes this unnecessary. The auxiliary valve F works on a seat G on the side of the valve-piston cylinder, being driven by the rod A and the rocker arm R from the main piston rod P. The length of its stroke can be changed by moving the pin B in the slot of the rocker arm, and the stroke of the main piston will thus be changed. The main valve is of the D form with a projection from the top which is engaged by the valve piston; the ports enter the cylinder, as in practically all direct-driven steam

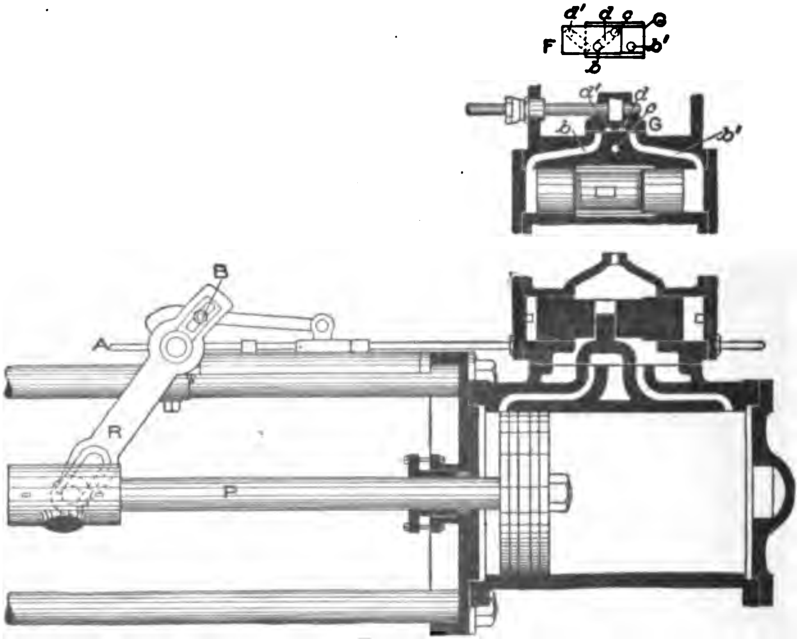


Fig. 53.

pumps, at some distance from the end; thus steam will be caught to form a cushion for the piston at the end of the stroke.

The auxiliary valve, Fig. 53, is made very short and is so arranged that its ports are open only when the main piston is near the end of its stroke. Ports *b b′* are for admission of live steam and port *c* is for exhaust. All three ports are covered by the valve except when near the end of its travel; then the groove *d* in the face of the valve connects ports *b* and *c* and allows the steam to exhaust from the left-hand end of the valve piston while port *b′* is uncovered and live steam is admitted to the right-hand end; or, at the opposite end of the stroke, groove *d′* connects *b′* and *c*, and *b* is uncovered. This construction prevents waste of steam if the valve piston becomes worn in its cylinder, since live steam is conducted to that cylinder only during the moment of moving the piston. The stroke of the pump can be easily regulated and the action is noiseless.

A valve with a peculiar cut-off action is used on the Blake pump. In this, as it stands in Fig. 54, steam is entering the head end of the cylinder through the ports E and H, and is exhausting from the crank end through ports H′, E′, K and M. As the main piston A nears the crank end of the stroke, the valve C is moved to the left by a system of levers similar to those on the Deane pump, which strike tappets on the rod P. When this happens the lug S on the valve C covers the auxiliary steam port N, and S′ uncovers the auxiliary steam port N′ (see plan view), while the auxiliary exhaust port Z is disconnected from X′ and connected to X, thus allowing steam to exhaust from the piston chamber at B′ and flow into that at B. The main valve D will be carried to the right by the supplemental piston and will connect port E to K, leaving E′ uncovered for the entrance of live steam, and the main piston will be forced towards the head end. It is not necessary to provide for positive mechanical driving of the main valve, as live steam can always enter one end or the other of both main and supplemental cylinders. The supplemental piston is fitted with rings so that it will take up its own wear, and prevent leakage into the exhaust, while the motion is such that the main valve has a slight lead on the main piston, thus admitting live steam to cushion the latter at the end of its stroke and prevent slamming. Also, in case the main valve sticks, the valve C moves far enough to admit sufficient steam past the end of the main valve for the cushioning effect. The supplemental piston cushions on the steam caught as it passes by the end of the port X or X'.

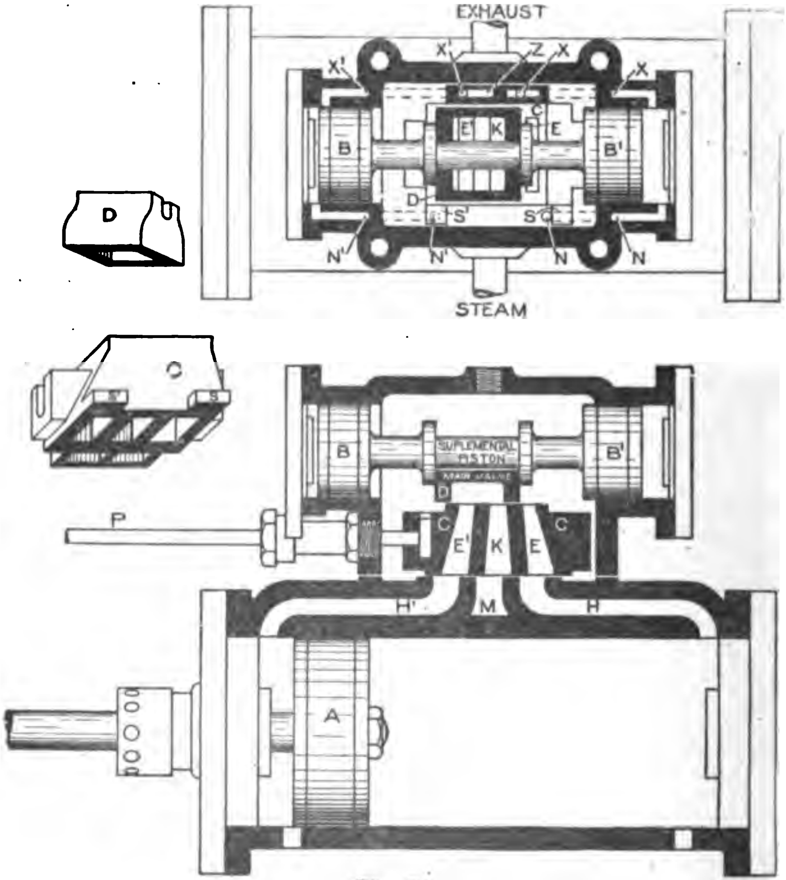


Fig. 54.

The Davidson pump, Fig. 55, has but one valve, operated by an auxiliary piston. The admission of steam to the ends of this piston is accomplished by the oscillating motion of the main valve itself. This motion is caused by a pin turned by a cam and oscillating lever driven from the pump rod. In the position shown, steam is being admitted to the crank end and exhausted from the head end; consequently the piston is moving towards the right. As it nears the end of the stroke, the cam C will rotate the pin D and place valve A in position to connect the port E' with the exhaust chamber while opening E to live steam as seen

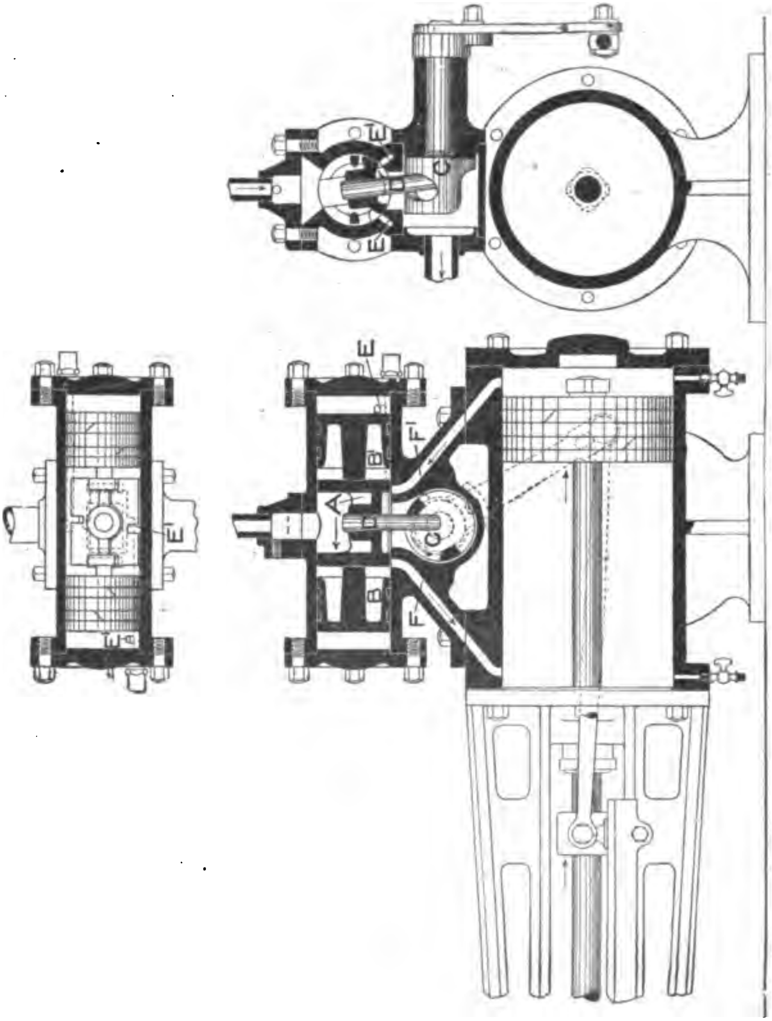


Fig. 55.

in the end view. The pin D will then be driven towards the left, carrying with it the valve A. Ports E' and F will be closed so that the main piston will be cushioned on the steam remaining in the cylinder.

The movement of valve A to this central position opens E′ to exhaust and E to live steam, and the piston B B′, carrying with it valve A, is forced to the left, opening port F to exhaust and F′ to live steam, and allowing the main piston to move towards the left When valve A is in a position to cover both ports F and F′, either E is open to steam and E′ to exhaust or vice versa, consequently the pump is always ready to start as soon as steam is turned on.

Various other forms of steam end have been used by different makers, but they work on practically the same principles as those described, being different only in details.

The duplex pump, as already explained, has two sets of cylinders side by side. The steam valve on one side is moved by means of a valve rod and lever connected to the piston rod on the other side, so that one side starts when the other has reached about three-quarters stroke. The steam valves are usually of the D form, having separate ports for admission and exhaust at each end as seen in Fig. 56. In this figure, valve B is moved by rod C and bell-crank lever D, the hidden end of which is driven from the farther piston rod by a device like H, while the farther valve is driven by the rocker-arm lever G F E. The double ports are used so that the piston may run over the exhaust port and close it before reaching the end of the stroke, thus getting a steam cushion, and yet live steam may have a chance to enter behind the piston even if it stops quite at the end of its possible motion. In large sizes (particularly in fire pumps) a cushion valve is placed

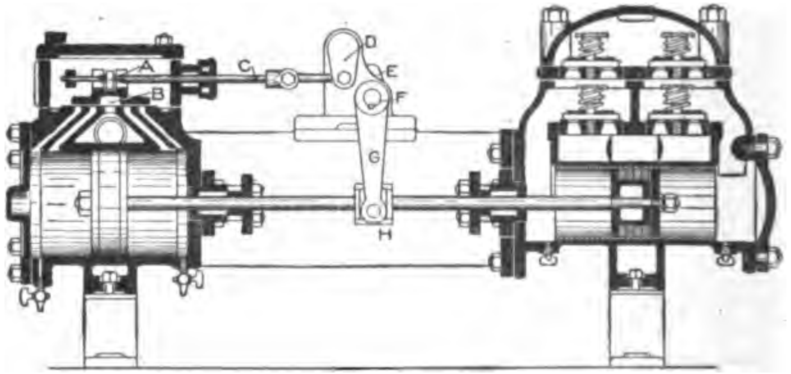


Fig. 56.

in a passage connecting the steam and exhaust ports, so that the amount of by-pass opening between them may be adjusted, thereby regulating the cushioning to any desired amount.

In order to secure the rest period at the end of each stroke, which is needed for quiet running and small slip, it is usual to introduce a lost-motion device in the valve-driving mechanism. In Fig. 56 this is the nut A, which has a thickness less than the space between the lugs on the valve, thus preventing the valve from being moved until the nut has traveled some distance. In larger sizes it is more common to use a single lug on the valve; the rod passes through this and the lost motion is provided for by adjusting nuts held fast by jam nuts, so that the valve lug will have play between them as seen in Fig. 57. For very large pumps and pumping engines, lost-motion links are usually placed in the end of the valve rod, as seen in Fig. 58; this arrangement has the advantage that the amount of lost motion can be adjusted without removing the steam-chest cover, or even, if desired, while the pump is in motion. It is, of course, somewhat more expensive to construct than the jam nuts shown in Fig. 57, hence is not often used for small pumps.

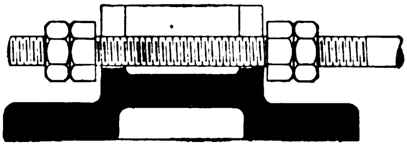


Fig. 57.

The amount of lost motion needed for any style of pump can be determined only by trial, but once fixed will be the same for all pumps of a given style and size. For pumps up to 10-inch stroke, $\frac{1}{8}$ to $\frac{3}{8}$ inch is usually allowed; for larger sizes the requirements call for $\frac{1}{2}$ to 1 inch.

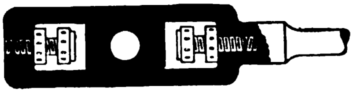


Fig. 58.

Compound Pumps. The simple steam pump must from the nature of its action take steam full stroke, hence has no possibility of using any of the expansive energy in the steam. In order to overcome this difficulty large pumps are often compounded, and usually with cylinders arranged tandem, as in Fig. 59, or with the smaller cylinder outside, as the designer may choose. Steam from the boiler enters the high-pressure steam-chest in the usual way; it then passes to the high-pressure cylinder; from there it exhausts through the side-pipe, seen at the back of the steam-chests in Fig. 59, to the low-pressure steam-chest; thence passes into the low-pressure cylinder and from it to the exhaust pipe. Unless a boiler pressure of more than 80 pounds is used, it is not advantageous to compound a pump run non-condensing, as the saving in steam will not pay for the increased cost. If a condenser is used, compounding may be introduced with profit for pressures as low as 50 pounds; but the added complication of condenser and many cylinders is inadvisable for pumps with low-pressure cylinders smaller than about 18 × 24 inches.

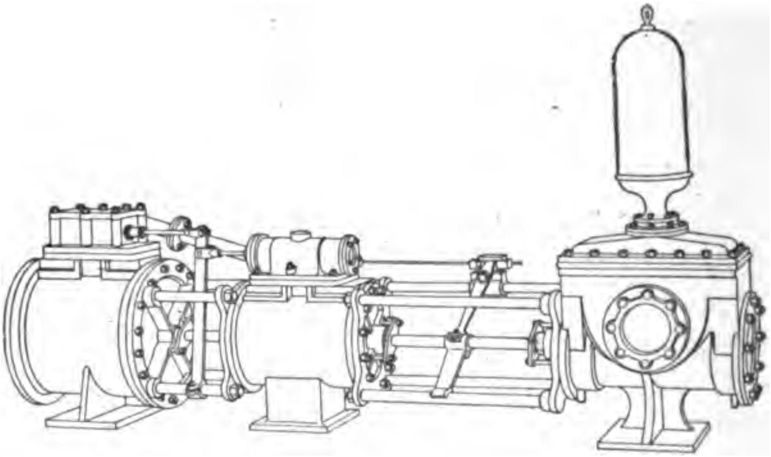


Fig. 59.

Compound pumps are used either single as in Fig. 60 with an auxiliary valve and valve piston, or duplex with the valves driven by the method described for duplex pumps. As the pistons are on the same rod, the valves must move at the same instant, and are usually mechanically connected and driven by the same valve piston or lever, as the case may be. The duplex compound is more often used than the single, and it is considered better practice to use a duplex compound with small cylinders than to use a single compound with larger ones, as the difference in cost will be more than balanced by the steadiness of running

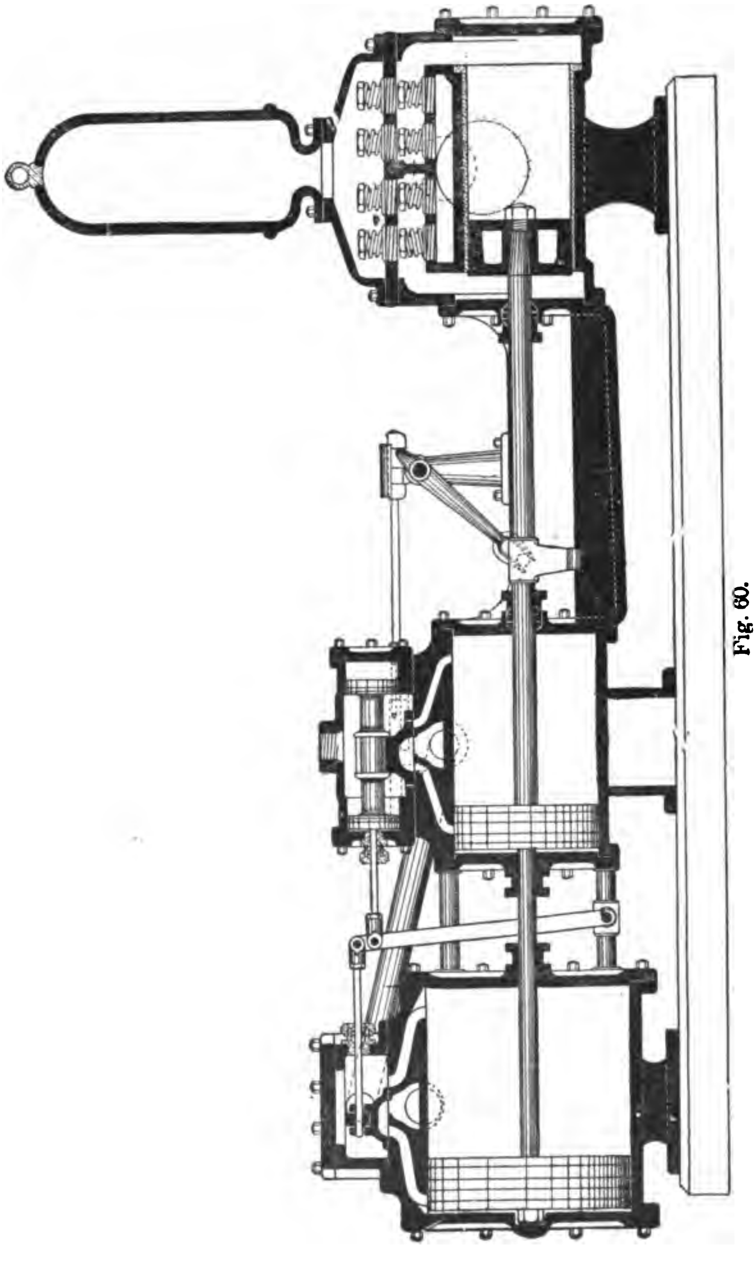


Fig. 60.

The high-pressure cylinder takes steam full stroke so that there is no expansion in it; at the opening of the high-pressure exhaust the pressure drops until it is equalized in the low-pressure steam-chest, side-pipe, high-pressure cylinder, and the clearance space of the low-pressure cylinder, the admission to which opens at the same instant as the exhaust from the high-pressure. On the return stroke of the pump, the steam expands in passing from the high-pressure cylinder to the low-pressure, the action being as indicated in the diagram, Fig. 61, and the nominal ratio of expansion between 2 and 3. In the diagram it is taken as 2.25, vertical distances being laid off to represent pressures and horizontal to represent volumes. The pressure in the passages beteen cylinders is a variable, but is taken as the value at the beginning of the low-pressure stroke. The back pressure in the low-pressure cylinder will be about 2 pounds above that at the outlet of the exhaust pipe, or 17 pounds for a non-condensing pump and 4 pounds for one run condensing.

The action illustrated in Fig. 61 is as follows: Assume that we are dealing with a pump whose high-pressure cylinder A is 16 inches in diameter by 18 inches stroke, and with clearance C 8 per cent of the piston displacement; and whose low-pressure cylinder B is 24 inches in diameter, the stroke being of course the same as for the high-pressure, and the clearance C 8 per cent of the low-pressure piston displacement. The intermediate space between cylinders, side-pipe and low-pressure steam-chest will be, in practice, about 0.35 the piston displacement of the high-pressure cylinder. Assume that the initial pressure is 85 pounds gauge or 100 pounds absolute and the back pressure in the low-pressure cylinder 4 pounds absolute. The line *a b* will represent the action during admission to the head end of the high-pressure cylinder, steam being taken at full pressure for the entire stroke. If we assume the volume of the piston displacement as 1, there will be 1.08 volumes of steam in the high-pressure cylinder and clearance. At the end of the stroke, the exhaust through the side-pipe into the low-pressure cylinder will be opened and the pressure will fall, the amount of the instantaneous drop depending on the pressure in the low-pressure steam-chest at the time the high-pressure exhaust opens. We shall see that this will be about 40 pounds, a value which can be safely assumed as a working basis. When the high-pressure exhaust opens, the admission to the low-pressure cylinder also opens, and the pressure will drop according to Boyle's law, so that it is equalized in the three compartments.

The volume of the low-pressure cylinder will be $1.5^2 \times 1 = 2.25$, and of its clearance $.08 \times 2.25 = 0.18$ volumes. Then the pressure after drop and equalization is found thus:

$$\begin{array}{rcr} 1.08 \times 100 & = & 108.00 \\ .35 \times 40 & = & 14.00 \\ .18 \times 4 & = & .72 \\ \hline 1.61 & & 122.72 \end{array} \qquad \frac{122.72}{1.61} = 76.2$$

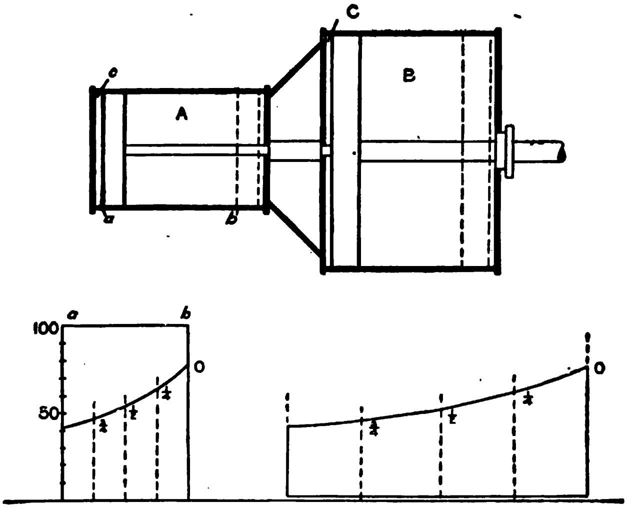


Fig. 61.

The pressure after equalization, 76.2 pounds, is found by multiplying each volume by the pressure existing within it and dividing the sum of the products by the sum of the volumes.

As the pistons return, the volume in the high-pressure cylinder decreases, that in the intermediate passages remains constant, and that in the low-pressure cylinder increases. The result will be an expansion according to the hyperbolic law so that the line showing the back pressure on the high-pressure piston, and that showing forward pressure on the low-pressure piston, will indicate the same pressure at each instant of the stroke, though the low-pressure volume will be 2.25 times the high pressure. At $\frac{1}{4}$ stroke the total volume will be

$.08 + .75 + .35 + .18 + (\frac{1}{4} \times 2.25) = 1.92$ volumes,

and the pressure $122.72 \div 1.92 = 64$ pounds.

Similarly at $\frac{1}{2}$ stroke the pressure will be 55 pounds,
at $\frac{3}{4}$ " " " " " 48.3 pounds,
at full " " " " " 42.9 pounds,

and so on for other points so that the whole expansion curve may be determined. In practice there will be a difference of about one pound between the back pressure in the small cylinder and the forward pressure in the large one, due to the friction of the steam in ports and side-pipe. On the next stroke, the head end of the high-pressure cylinder will take in a fresh charge of steam while the crank end of the low-pressure will exhaust its steam into the condenser. The crank end of the small cylinder and the head end of the large one act together in the same manner as described above, so that the pump is double-acting.

We are indebted to a paper on "Power of Compound Pumping Engines," by John W. Hill, published in *Engineering News*, for the proportions given in Table VIII:

TABLE VIII.

Ratio diameter l. p. cylinder to h. p. "	1.50	1.60	2.00
High-pressure cylinder volume taken as	1.00	1.00	1.00
High-pressure clearance volume	.063	.063	.06
Intermediate space volume	.339	.339	1.12
Low-pressure cylinder volume	2.25	2.56	4.00
" " clearance "	.112	.128	.204
No. expansions intermediate chamber	1.319	1.439	2.250
No. expansions l. p. cylinder	1.825	2.020	2.258
No. expansions, total	2.400	2.907	5.081
Ratio Mean Effective pressure to (initial pressure minus back pressure)	0.729	0.682	0.466

Setting the valves on a pump is for the most part a simple operation, and requires only that the valves shall be adjusted, usually by trial, until the pump makes the longest stroke possible without striking the heads, and reverses evenly at the ends of the stroke.

The fly-wheel pump is adjusted by moving the eccentric on the shaft and the valve on its stem until steady running is secured, the same as for any slide-valve engine. The Knowles pump has the stroke lengthened by lowering the roll E, Fig. 50, and shortened by raising it; equalization of reversal is effected by lengthening or shortening the link L as may be required.

Adjustment of the Deane pump is made entirely by moving the tappets on the valve rod; the block A should be clamped to the piston rod in such position that when the piston is at mid-stroke the link B will be vertical. With the valve in mid-position the tappets should be placed so that they are equidistant from each end of the sliding collar, and then adjusted by trial until the working is satisfactory. If the piston strikes at either end, move the tappet at that end towards the collar; if reversal comes too soon at one end, move the tappet away from the sleeve at that end.

For the Dean Bros.' pump the same directions apply as to the Deane, but the stroke may be changed by moving the bolt B, Fig. 53, in its slot without changing the tappets.

The adjustment of the Blake valve is also the same as that for the Dean Bros.', but the tappets on the valve rod should be so set that the valve will have a little lead and open before the main piston reaches the end of its stroke. It has this lead by virtue of the action previously explained unless it is set to be very late in action.

In the Davidson pump, Fig. 55, the valve is adjusted for length of stroke by moving the bolt in the slotted end of the oscillating arm; shortening the leverage shortens the stroke and vice versa. To equalize the reversal, the sleeve which is clamped to the piston rod may be shifted towards the end at which it is desired to quicken the reversal, or a slight adjustment may be made at the point where the oscillating lever is made fast to the rock-shaft which drives the cam C.

For a single compound pump the valves are adjusted in the same way as for a simple pump of the same make. There is an adjustment in the connection between high- and low- pressure valves, and this must be set, with the steam-chest covers off. so that the main valves open at the same time.

All the valves of the duplex pump are set square so that, with

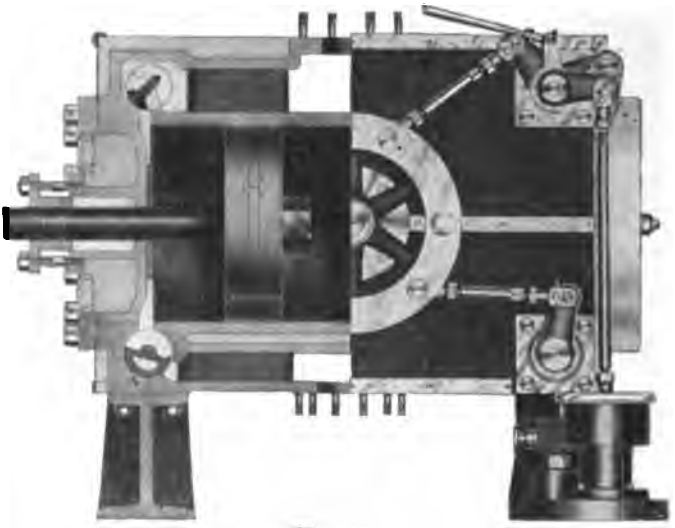


Fig. 62.

Fulton Corliss Cylinder.

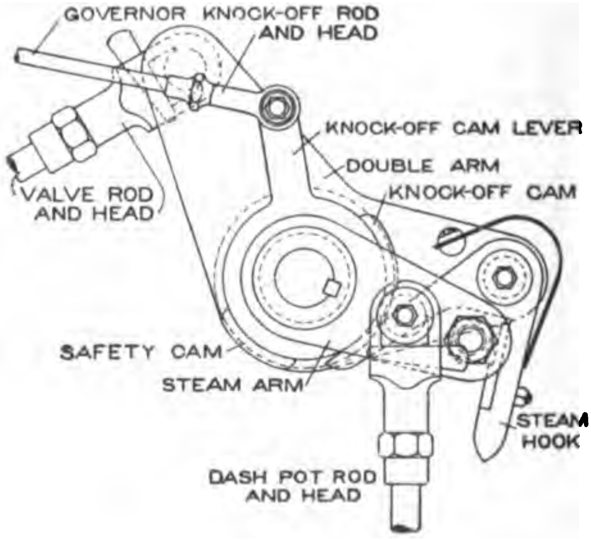


Fig. 63.

both pistons at the middle of the stroke, the valves will be in mid-position, the rocker-arm levers will be vertical and the lost motion will have equal play at each end. The amount of lost motion needed, where it is adjustable, can be determined only by trial, but generally should be as great as possible without having the pistons strike the heads. This will give a long stroke.

For the large pumping engines used in city water-works, the steam ends are designed the same as for any steam engine; and fly-wheels, main shafts and eccentrics are provided for steadying the motion, allowing of expansive working and operating the valves. For such engines the Corliss valves and valve motion have been most commonly used in the United States.

The arrangement of the motion and the action of the valves is shown in Fig. 62 The wrist plate is moved by the eccentric through a reach-rod, rock-shaft and eccentric rod, and from it links run to cranks which turn the rotating valves in their seats. The exhaust valves are positively driven, but the admission valves are so arranged as to be disconnected from the control of the link at some point in the stroke (depending on the position of the governor) by a trip motion, one form of which is shown in Fig. 63.

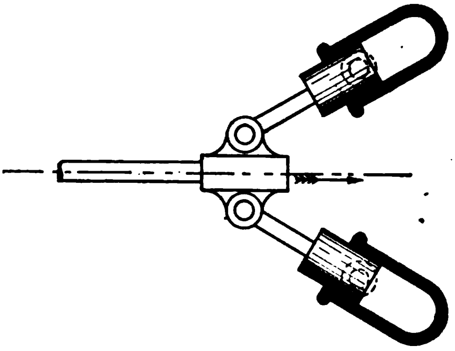


Fig. 64.

Another device used to allow of expansive working of steam in large pumps is the hydraulic compensator, Fig. 64. The steam must work against the pressure of the pistons up to half stroke, and is assisted by it beyond that point, so that cut-off may take place at half stroke, or later, and the energy of expansion be used beyond that point. With this device, there is no shaft or eccentric, and valves similar to those described for duplex pumps are used.

ERECTION AND PIPING.

The location of a pump should be chosen with two principal objects in view: To have the pump itself convenient for running, and accessible for repairs or adjustment; and to keep the piping as short and direct as possible. For a pump to which the liquid is raised by suction there is the limitation that it must not be placed more than 25 feet, and should not be more than 20 feet, above the source of supply; but aside from this, a matter of first importance is to have the machine where it will naturally be kept in good condition. Too often a pump is placed in a dark corner where it is never seen, seldom visited, and always neglected; it soon becomes dirty and leaky, decreasing its efficiency and shortening its life. The second point with respect to piping is often controlled by the layout of other apparatus quite as much as by the position of the pump itself; yet by careful study of the conditions it is often possible to find one place better than others for the pump. It is of more importance to avoid bends and elbows in water piping than in that for steam, because steam makes sharp turns with less friction loss than does water. If hot water is to be handled, the water should flow to the pump by gravity or under pressure. Otherwise the water will turn into vapor under the suction force and the pump will draw in either vapor alone or a mixture of vapor and water.

It is well to place a boiler feed pump in such a position that the gauge glass can be seen when standing at the pump, but this is not absolutely essential, and should be sacrificed if any gain in arrangement of piping or convenience of attendance can be secured thereby. If a condenser air-pump is so placed that the water from the condenser flows to it by gravity, it is possible to maintain a better vacuum in the condenser.

Wherever the pump may be located, a substantial foundation should be provided if the pump is of large size, especially if it is to be run at high speed. It is often sufficient to fasten a small pump to the floor or to heavy brackets secured to a wall, but a pump larger than a 4 × 6 should have a separate foundation. In designing the foundation, remember that the object is not so much to hold the pump up as to hold it down, to keep it from vibrating.

It is, therefore, better to have a foundation deep and narrow than broad and shallow, unless the pump is large and the soil very sandy.

The pump should be well bolted to the foundations in order to prevent vibration, as such movement not only is communicated to the pipe and thence to the building, but tends to loosen the joints in the piping and the pump itself. Absolute rigidity on the foundations should be secured at any cost.

The material may be stone, brick, or concrete, preferably the last. Stone is expensive and difficult to work; brick is liable, unless carefully laid with cement mortar, to be loose and lack compactness; while concrete is easily put down, is inexpensive and has all the solidity of stone. It should be made of good cement mortar, two parts sand to one Portland cement, mixed with broken stone not over 2 inches in longest diameter, in equal parts of stone and mortar. The concrete should be well compacted into a mould the shape of the foundation, the bolts being built in with plate-iron washers on the heads; the concrete should be deposited in layers about 6 inches deep and well rammed, a second layer being added before the upper surface has hardened. This process is repeated until the foundation is completed. It will generally be sufficient to finish the top with a surface of cement mortar carefully leveled and allowed to harden before setting the pump, but sometimes a cast-iron base-plate is used, and this gives a somewhat neater appearance.

As previously stated, the piping should be as short and direct as possible. In large work, the water pipe should have long-bend elbows and tees, and gate valves should be used to reduce the friction. Each pipe should be pitched throughout its length to one point so that it may be drained to avoid freezing; a drain-cock should be placed at the lowest point to remove the water. For water piping, it is well to use galvanized or brass pipe to avoid pitting or corrosion. Covering the pipes which carry cold water will prevent sweating and the consequent unpleasant dripping.

If the plant is one where a shut-down would be serious, a duplicate system should be installed, pump, piping and all; for any plant it is well to provide an injector as relay, in case the boiler feed-pump will not work. In some cases duplicate piping is installed, but this seems hardly necessary, as piping is not likely to get out of order if well taken care of.

Wherever a long column of water is to be moved in either suction or delivery pipes, it is well to place a check valve near the lower end of the column to resist any tendency of the water to back up when the pump reverses or shuts down. In the suction system this valve would be placed on the inlet end of the suction pipe and is known as a foot valve, Fig. 65. For the delivery pipe, it is well to use a check valve near the outlet from the pump and another near the end of the pipe, especially if pumping against high pressure. The check valves may be of either flap or disc type, but if of the latter they should have ample area so that the double turn made by the current of water will not cause great loss of head. The flap valve is shown in Fig. 66 and the disc form in Fig. 67.

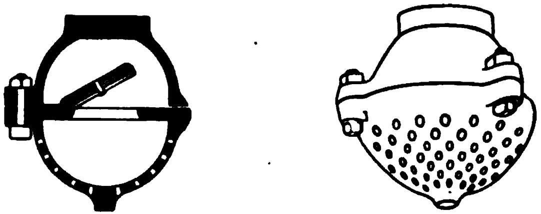


Fig. 65.

For all pumps which are to handle water from ponds, rivers or other sources where sticks, leaves, or any form of rubbish is likely to collect, a strainer as shown in Fig. 65 should be placed on the end of the inlet pipe. The combined area of the openings into the strainer should be 3 to 4 times the area of the pipe. As rubbish quickly collects on a horizontal strainer, the surface should be either slanting or vertical, and should be so designed that the screen may be easily cleansed. The straining surface should be fine or coarse in mesh according to the material to be screened, and made of woven wire or perforated metal as may be most con-

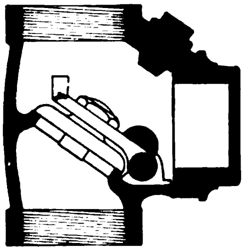


Fig. 66.

venient. Often the foot valve and strainer are combined into a single piece. If the lower end of the suction pipe is not accessible for cleaning, and if the debris is of such nature that it is likely to clog the openings, it will be better to use a design given by Barr (see Fig. 68) placed near the pump.

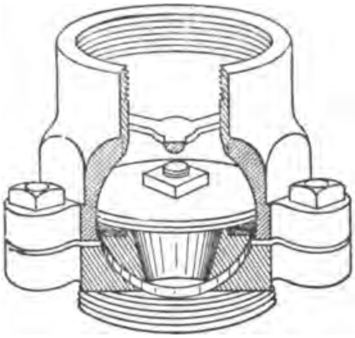


Fig. 67.

The sizes of pipe needed for the suction and delivery are determined by the pump maker and the pipes should never be made smaller than these; if the runs are long, the pipes may well be made larger.

The velocity allowable is, of course, the point which determines these diameters. For the suction, this is usually taken at 200 feet per minute or less; for the delivery pipe it may be 400 feet per minute or less.

The suction pipe should be of the same size throughout in order to avoid eddies and changes of velocity. Where the pipe is larger than the pump connection, the reduction should be made by a conical pipe with an easy taper, placed next the pump. The greatest care should be taken to see that all joints in the suction system are absolutely tight, as even a small leak greatly reduces the capacity and efficiency. The diameters of pipe suitable for direct-acting pumps are given in Table IX, the suction velocity being allowed at 150, and delivering velocity at 300 feet per minute.

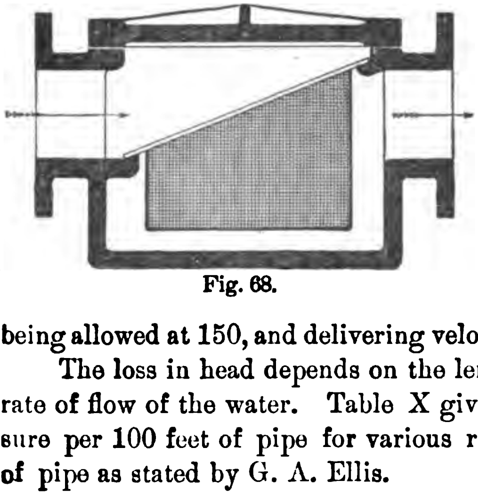


Fig. 68.

The loss in head depends on the length of the piping and the rate of flow of the water. Table X gives the loss in pounds pressure per 100 feet of pipe for various rates of discharge and sizes of pipe as stated by G. A. Ellis.

The loss of head from elbows and valves depends also on the

TABLE IX.

SIZES OF SUCTION AND DELIVERY PIPES.

Diameter Water Cylinder.	Diameter Suction Pipe.		Diameter Delivery Pipe.	
	Single Pump.	Duplex Pump.	Single Pump.	Duplex Pump.
4 inches	2½ inches	3½ inches	1¾ inches	2½ inches
5 “	3 “	4 “	2 “	3 “
6 “	3½ “	5 “	2½ “	3½ “
7 “	4½ “	6 “	3 “	4 “
8 “	5 “	7 “	3½ “	4½ “
9 “	6 “	8 “	4½ “	6 “
10 “	6 “	8 “	4½ “	6 “
12 “	7 “	10 “	5 “	7 “
14 “	9 “	12 “	6 “	8 “
16 “	10 “	14 “	7 “	9 “
18 “	12 “	16 “	9 “	12 “
20 “	12 “	17 “	9 “	12 “

Suction velocity, 150 feet per minute.
Delivery velocity, 300 feet per minute.
Piston speed, 100 feet per minute.

rate of flow, and is most conveniently referred to the length of pipe which would result in the same loss.

The resistance to flow of water in pipes due to bends, elbows, tees, etc., is stated by Foster in his “Electrical Engineer's Pocket Book,” to be expressed by the equation:

$$P = F\frac{v^2}{64.4}$$

in which P is the loss in pressure in pounds per square inch, v the velocity of flow in feet per second, and F the coefficient of friction, which varies with the angle of the bend according to the following table:

Angle.	20°	45°	60°	90°	120°	135°
F	.020	.079	.158	.426	.806	.940

A globe valve will produce the same loss of head as two 90-degree bends, and a gate valve a loss equal to that from a 45-degree bend.

If water is known to contain lime or magnesia, it is certain that pipes will fill up more or less from the deposit of scale, and allowance should be made for this in the first place by using extra large pipe.

In choosing the size of steam pipe the same general principle applies; the exhaust pipe, like the suction, should be short, direct, and of ample cross-section. A velocity of 6,000 feet per minute is allowable in the steam pipe and 4,000 in the exhaust.

Care should be taken to have all piping which carries steam pitched away from the pump to avoid the collection of the water of condensation in the steam cylinder, and drips should be provided wherever there is any chance for water to collect.

The use of air chambers has already been discussed. They are usually a good investment if high speed or long pipe runs are to be used. The gain in durability and saving of repairs to the pump and piping system will more than pay the interest on the small cost of ample air chambers.

TABLE X.

FRICTION OF WATER IN PIPES.

Friction loss, in pounds pressure per square inch, for each 100 feet of length of different sizes of clean iron pipe discharging given quantities of water per minute.

G. A Ellis, C. E.

Gallons per Minute	SIZES OF PIPES—INSIDE DIAMETER														
	¾ in.	1 in.	1¼ in.	1 in.	2 in.	2½ in.	3 in.	4 in.	6 in.	8in.	10 in.	12 in.	14 in.	16 in.	18 in.
5	3.3	0.84	0.31	0.12	0.05	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....
10	13.0	3.16	1.05	0.47	0.12	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....
15	28.7	6.98	2.38	0.97	0.30	0.11	.....	.....	.....	.....	.....	.....	.....	.....	.....
20	50.4	12.3	4.07	1.66	0.42	0.15	.....	.....	.....	.....	.....	.....	.....	.....	.....
25	78.0	19.0	6.40	2.62	0.51	0.21	0.10	.....	.....	.....	.....	.....	.....	.....	.....
30	.....	27.5	9.15	3.75	0.91	0.33	0.11	.....	.....	.....	.....	.....	.....	.....	.....
35	.....	37.0	12.4	5.05	1.20	0.45	0.17	.....	.....	.....	.....	.....	.....	.....	.....
40	.....	48.0	16.1	6.52	1.60	0.52	0.22	.....	.....	.....	.....	.....	.....	.....	.....
45	.....	.....	20.2	8.15	2.00	0.65	0.28	.....	.....	.....	.....	.....	.....	.....	.....
50	.....	.....	24.9	10.0	2.44	0.81	0.35	0.09	.....	.....	.....	.....	.....	.....	.....
75	.....	.....	56.1	22.4	5.32	1.80	0.74	0.17	.....	.....	.....	.....	.....	.....	.....
100	.....	.....	.....	39.0	9.46	3.20	1.31	0.33	0.05	.....	.....	.....	.....	.....	.....
125	.....	.....	.....	.....	14.9	4.89	1.99	0.53	.....	.....	.....	.....	.....	.....	.....
150	.....	.....	.....	.....	21.2	7.0	2.85	0.69	0.10	.....	.....	.....	.....	.....	.....
175	.....	.....	.....	.....	28.1	9.46	3.85	1.00	.....	.....	.....	.....	.....	.....	.....
200	.....	.....	.....	.....	37.5	12.47	5.02	1.22	0.17	.....	.....	.....	.....	.....	.....
250	.....	.....	.....	.....	.....	19.66	7.76	1.89	0.26	0.07	0.03	0.01	.....	.....	.....
300	.....	.....	.....	.....	.....	28.06	11.2	2.66	0.37	0.09	0.04	.....	.....	.....	.....
350	.....	.....	.....	.....	.....	.....	15.2	3.65	0.50	0.12	0.05	0.02	.....	.....	.....
400	.....	.....	.....	.....	.....	.....	19.5	4.73	0.65	0.16	0.06	.....	.....	.....	.....
450	.....	.....	.....	.....	.....	.....	25.0	6.01	0.81	0.20	0.07	0.03	.....	.....	.....
500	.....	.....	.....	.....	.....	.....	30.8	7.43	0.96	0.25	0.09	0.04	0.017	0.009	0.005
750	.....	.....	.....	.....	.....	.....	.....	.....	2.21	0.53	0.18	0.08	.....	.....	.....
1000	.....	.....	.....	.....	.....	.....	.....	.....	3.88	0.94	0.32	0.13	0.062	0.036	0.020
1250	.....	.....	.....	.....	.....	.....	.....	.....	.....	1.46	0.49	0.20	.....	.....	.....
1500	.....	.....	.....	.....	.....	.....	.....	.....	.....	2.09	0.70	0.29	0.135	0.071	0.040
1750	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	0.95	0.38	.....	.....	.....
2000	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	1.23	0.49	0.234	0.123	0.071
2250	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	0.63	.....	.....	.....
2500	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	0.77	0.362	0.188	0.107
3000	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	1.11	0.515	0.267	0.150
3500	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	0.697	0.365	0.204
4000	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	0.910	0.472	0.263
4500	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	0.593	0.333
5000	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	0.730	0.408
Comparative Discharging Power of Pipes	$\sqrt{d^5}$	1	1.75	2.76	5.66	9.88	15.59	32.	88.2	181	316.2	498.8	733.4	1024.	1375.

Care should be taken, however, that no pockets are formed in the piping where air may collect, as the air cushion thus formed will serve no useful purpose, but will reduce both the capacity and the efficiency of the pump. The suction pipe should have a continuous rise from the source of supply up to the pump; and if an inverted U loop must be formed in the delivery piping it should have a pet cock inserted at the highest point so that whatever air collects may escape.

CARE.

After the valves of a pump are properly adjusted, the three things which ordinarily require care are: The lubrication, the packing, and the draining of cylinders. If these matters are carefully attended to, the pump will cause very little trouble.

For the smaller sizes the steam end is generally lubricated by means of a grease cup, which is filled with some form of tallow compound. As the heat of the steam melts this compound gradually, it flows into the steam-chest and is carried by the steam to the cylinder. For larger sizes, a regular cylinder oiler is used, or sometimes an oil pump driven by a lever from the main piston rod. In large plants, the pumps as well as the engines are fed from a central tank into which the oil is forced under pressure by a single large oil pump, and whence it descends by gravity to the various cylinders and passes through sight-feed oilers.

The stuffing box on the steam end usually gets sufficient lubrication from the cylinder, and the one on the water end gets water enough except when the packing is set up hard; then a little machine oil with graphite in suspension will help.

In the case of the steam-cylinder as for any other engine, the less oil used, so long as the piston works quietly, the better.

Flake graphite put into cylinder oil usually settles to the bottom of the cup, but if blown into the steam pipe so as to be carried along by the steam, it will work into the crevices in valves and piston rings and aid materially in reducing the oil required. Also if sifted over the packing when filling the stuffing boxes it will reduce the friction considerably.

For the bearings of valve motions, machine oil is, of course, used. These parts need the same care as any other machine bearings.

The packing in the water piston wears but slowly; nevertheless it should be regularly inspected to make sure that there are no leaks, as they would seriously impair the economy of the pump. Once in two months is not too often to examine the water pistons, and oftener should be the rule if there is any reason to suspect trouble. In repacking, or in tightening up either piston or stuffing boxes, there should be as little pressure as possible, above the limit to prevent leakage.

The piston ring packing of the steam end will wear for years if properly adjusted and lubricated. An inspection once a year is sufficient unless suspicious action in the cylinder seems to call for an investigation.

The stuffing boxes should be kept tight, but not screwed up so as to bind. It pays to be rather generous both in the amount of packing used in a stuffing box and in frequency of renewals; the former because a long bearing between rod and packing will keep tight with less pressure than a short one; the latter because old, hardened packing requires a heavy pressure to force it to a tight joint, and results in a large amount of energy wasted in overcoming friction. For stuffing boxes, any good square packing will answer, but the one on the water end is better filled with some form having a rubber compound for its main body, while that on the steam end works better with a flax packing, as the steam soon kills the rubber. Stuffing boxes should be refilled as often as twice a year, if the pump is in constant service.

The valves in the steam end have a sliding bearing, and will ordinarily wear to a true and tight joint. At the time of the yearly inspection, the head should be removed, steam turned on, and the valve worked back and forth by hand to make sure that no steam passes into the cylinder except when the admission valve is properly open. If the valves leak, they must be scraped to a bearing. Usually the valve face is scraped accurately by using a surface plate, and the seat is then scraped to fit the valve.

The valves in the water end may be reseated by grinding the seats and faces with tools made for the purpose, if both the valves and seats are of metal. If the valves have a bearing surface of composition, as is now the usual practice, the composition disc will wear rather than the seat, and the disc may be renewed easily and at slight expense. Some makers use discs of soft alloy which are more durable than composition and also have its advantages.

In discussing the subject of capacity, the matter of speed was taken up. Manufacturers conventionally rate their pumps at 100 feet per minute, but this is not a good basis, and sixty double strokes seem to be more logical for computation. The object is to reduce slip and prevent pounding; hence, if special devices are used for opening and closing the water valves and to prevent slamming at the end of the stroke, there is no reason why a speed may not be used approaching that of power engines. The slip occurs almost entirely at the ends of the stroke and during the seating of the valves, so that it is always well to use a long stroke even though the diameter be somewhat small.

If mechanically operated valves together with high speed are used, ample air chambers should be placed on suction and delivery systems, as otherwise there is likely to be water-hammer due to the sudden stoppage of the column of fluid in the pipes when the valves close.

There is no difficulty with the steam end in using high speeds provided the steam is kept free from water. This necessitates, in the case of large pumps, a separator in the steam pipe just above the pump, to remove all water and ensure the passage of none but dry steam.

It is impossible to forewarn against all difficulties which may arise in running a steam pump, because it is always "something different" which happens, but trouble can often be traced to certain common faults. Defective valves in the water end and stoppage in the suction pipe are the probable causes for irregular working of a single pump. If the pump slams on one stroke and is steady on the other, it may be that the discharge valves are stuck open on one end either by friction or the lodgement of some substance on the seat. Often a jar with a hammer will remedy this defect, but it is advisable to take off the valve-chamber cover to find the reason for the sticking. If the slamming is on both strokes, it is generally due to stoppage in the suction pipe, or, in the case of a pump newly erected, it may be that the suction is too small. If the latter is the case, slowing down will stop it; but if there is stoppage, the pump will slam at all speeds. If the suction is small, the addition of a suction air chamber will sometimes be beneficial. Slamming may also be due to a leak in the suction system in either valves or joints, or to a leak in the piston packing. Occasionally the springs on the inlet valves are too strong, though this is seldom the case. When starting up, air in the pump may cause it to slam, the remedy being, of course, to prime the pump and suction pipe by pouring in water, and to make sure, by opening the air cock on top of the water end, that all air is forced out of the valve chamber.

If a pump sticks at the end of the stroke, it is due to friction or improper valve setting. In the former case, relieving the pressure on the nuts which set up the glands to the stuffing boxes, until there is just enough pressure to prevent leakage, will overcome the difficulty. In the latter case the valve motion should be so adjusted as to act earlier in the stroke, but it is best to keep the stroke of the pump as long as possible in order to reduce the loss from clearance in the steam cylinder.

In starting up, particularly in cold weather, there will be considerable condensation in the cylinders, and water will form rapidly. This must be given a chance to work its way out through the drips which should be left open until the pump runs free and without sign of water in the steam; the warming up should be done at slow speed.

If a cylinder oiler is used it should be opened up a sufficient time before the pump is to be started, so that it may be ready to act immediately when the pump starts, as the lubrication is needed when the cylinder is cold, even more than at any other time. It is well to have a hand-forcing oil pump connected to the steam pipe of large pumps unless the feed of oil is by a positively-driven pump, so that in starting up, or in case of emergency, a supply of oil may be ensured.

Before starting, the suction and outlet valves should be inspected to make sure that both are open. If a start is made with these closed, it is likely to bring a pressure on the systems which will open the joints.

If the pump is a large one and is run condensing—that is, exhausting into a condenser—the condenser should be put in operation first by starting the flow of cooling water and the air pump.

When these are both working well and a good vacuum has been established, the main pump may be started.

In closing down, the lubricators may be closed a little before the time to stop. The drips on the steam cylinder should be opened after stopping in order to carry off the condensation from any steam which may remain. The drips on the water cylinder need not be opened unless there is danger of freezing, in which case the whole water system should be drained. For this purpose the piping should all pitch toward the pump so that the water may all be drawn off at that point.

In the duplex type of pump, unless the packing be adjusted with even pressure on both sides, the side on which the tighter adjustment is made is liable to "short stroke;" in fact this is usually the trouble with a pump which goes "lame" on one side. If the short stroking is on one end only, it is probably due to poor setting of the valve-motion tappets. Short stroking may also be due to tight packing on the water piston, in which case it can be remedied by taking out the packing and cutting off a little. This is likely to occur only with pistons packed with square packing, as those having the cup leather packing or piston rings adjust their own pressure between piston and cylinder.

In setting up a new pump, it is important to blow out all pipes before making the connections, in order to make sure that no chips or dirt get into the pump. Unions should be provided on each pipe near the pump, so that, in case of suspected stoppage of the pipe it can be readily inspected.

TESTING.

The power used by a pump is usually so small an item in the running expense of a plant, that a test is considered unnecessary; and also the steam exhausted is often used for heating feed water or for some industrial process. If this is possible it is usually an economical way of running; but in plants where many pumps are needed, and where heating feed water is the only use for the exhaust steam, more steam will be available than can be used to good advantage; this is especially true if the main engines run non-condensing. An economical pump is then of great importance.

Duty. In order to determine the good or bad performance of a pump, it is necessary to test it for efficiency and slip, the test being known as a "duty trial." The duty of a pump, as the term descended to us from the days of Watt's early pumping engines, was the number of foot pounds of work produced by 100 pounds of coal. This is a convenient basis for comparison of engines, but is not accurate, as coal varies so much in heat value; also this method of reckoning involves the efficiency of the boiler as well as that of the pump. As an attempt to eliminate the latter source of error in making comparisons, a conventional assumption has been made of 10 pounds of steam evaporated per pound of coal; but this is really only the substitution of one error for another, for a pound of steam when measured in heat units is by no means a constant, and a pound of coal seldom does evaporate 10 pounds of steam in actual boiler performance. The more logical method of comparison is by the duty per 1,000,000 heat units furnished to the pump, a basis proposed by a committee of the American Society of Mechanical Engineers appointed to formulate a code for conducting such tests.

The duty of the pump is found by measuring the quantity of water delivered and the height through which it is lifted, or its pressure equivalent. The coal or steam used must also be measured. Then

$$\text{Duty} = \frac{Q H \times 100}{C}$$

in which Q = pounds of water delivered,
H = head against which the pump works, both suction and forcing, and
C = pounds of coal burned.

Or on the new basis.

$$\text{Duty} = \frac{Q H \times 1{,}000{,}000}{B T U}$$

where Q and H are as before, and $B T U$ is the **heat** units in the coal or steam, whichever is measured.

An inaccurate method of computing duty is sometimes used, which is based on the area of the water piston or plunger and its travel; but this takes no account of the slip, which may in small pumps be as much as 20 per cent; hence it is to be condemned.

The slip is found by comparing the water actually delivered with the total piston or plunger displacement, for the time of the test. The difference, measured as a per cent of the piston displacement, is the slip.

The water delivered is measured either by weighing, by the use of calibrated tanks, by a weir, or by some form of meter. One of the first two methods is best for small pumps, and the meter is practically the only means available for large ones where the delivery is into a closed pipe system. For the weight or tank method, the arrangement of Fig. 69 is used, one tank being filled while the other is being emptied, through a quick opening gate valve. If the water is weighed, a certain amount as nearly as may be is run into the tank and the exact weight is caught after the valve is closed and the water is flowing into the other tank. For calibrated tanks, the filling is done nearly to a set mark, at which the capacity of the tank is known, and the exact level is found by dipping from the second tank. The weir can be used only where the delivery is into an open vessel, as the current of water must be made to flow over an open notch as in Fig. 70. The amount of water which passes this notch evidently depends on its length, and on the head of water above the sill. There is a certain contraction as the water enters the notch; but if the edges are beveled to a sharp edge up stream as shown, this will be slight. The depth of the notch should be not over $\frac{1}{2}$ the length, and is better made considerably less, say from $\frac{1}{10}$ to $\frac{1}{4}$. The over-fall below the notch should be at least twice the depth of the notch. The head of water over the sill should be measured at a point some distance back of the notch in order to get a quiet, even surface.

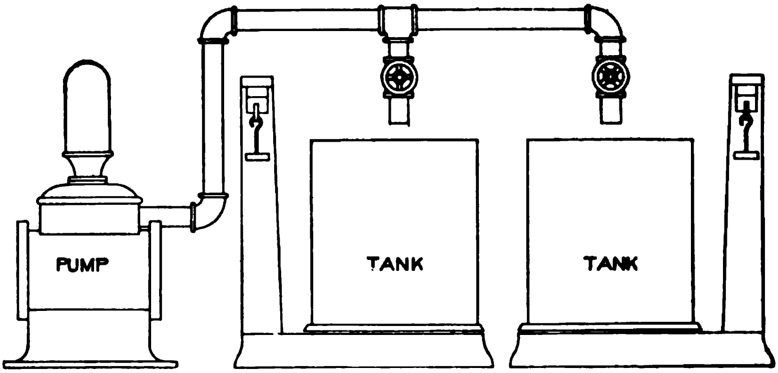


Fig. 69.

Any method of measuring the head will answer which gives it accurately; but a hook gauge, such as shown in Fig. 71, will give the best results. The reading of the gauge should be taken when set so that the point just breaks the surface as it is brought up from below, when no water is flowing over the notch. This gives the height of the sill from which to calculate. When the water is flowing from the pump, a reading from the gauge is again taken as the point just breaks the surface. The difference between the two readings is the head above the sill.

The flow, if there were no end contraction, would be $Q = b\,v\,h$ where Q is the cubic feet per second, b the length, h the head of water above the sill, and v the velocity in feet per second. But $v = \sqrt{2\,g\,\frac{h}{2}}$ the same as for falling bodies, $\frac{h}{2}$ being used because it is the head of center of flow. It has been found by experiment that there is a contraction at the ends and bottom of an opening, due to the in-rush of the water from all sides, and that the flow will be about .62 the theoretical amount; hence

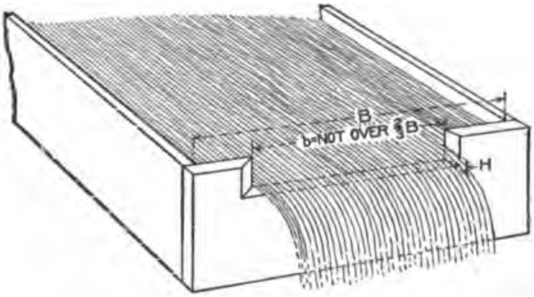


Fig. 70.

$$Q = .62\,bh\sqrt{2g\,\frac{h}{2}}$$
$$= .62\,bh\sqrt{gh}$$
$$= .62\,bh^{\frac{3}{2}}\,g^{\frac{1}{2}}$$

but g is the acceleration due to the force of gravity and is equal to 32.16; hence $.62 \times g^{\frac{1}{2}} = 3.52$ and $Q = 3.52\,bh^{\frac{3}{2}}$.

It is found, however, that for accuracy the formula must be modified to take account of the depth of the water, and also that the coefficient is too large. Smith gives, in his Hydraulics, the equation

$$Q = 3.29\left(b+\frac{h}{7}\right)h^{\frac{3}{2}}$$

which is accurate for weirs for depth h, from 6 inches to 2 feet and with length b, not less than $3h$. For large quantities of water which must be delivered under pressure, the Venturi meter is the most accurate means of measurement. This is a patented device manufactured by the Builders Iron Foundry of Providence, R. I., and registers the flow by means of a recording mechanism driven by clockwork. The next best device is some form of rotary water meter, carefully calibrated. These will work well with cold water, but are not to be relied upon with hot water as the varying temperatures affect the readings appreciably.

For methods of testing large pumping engines, the student is referred to the report of the committee of the American Society of Mechanical Engineers, Vol. XI of the Transactions, which can be obtained from the Society in pamphlet form at nominal cost.

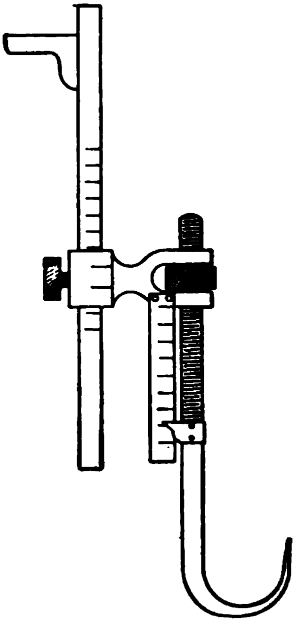


Fig. 71.

The following discussion will be confined to small pumps :

For a power-driven pump it is necessary to measure the power supplied. If belt-driven, this can best be done by means of a transmission dynamometer, which measures the pull on the belt. Then this pull in pounds times the speed of the belt in feet per minute equals the foot pounds per minute; and that quantity divided by 33,000 gives the horse power. The work performed per minute will be the weight of water pumped per minute times the distance through which the water is raised as indicated by gauges on the suction and delivery pipes near the pump. These heads can be derived from the ordinary pressure gauge readings by the table on page 9. The work per minute divided by 33,000 gives the delivered horse power. For a motor-driven pump the power applied can readily be obtained by electrical measurements. Measure the voltage and amperes of current with the pump carrying its load and then Voltage $\times \frac{\text{Current}}{746} =$ Horse power. From an efficiency curve of the motor as supplied by the makers, or obtained by a dynamometer test of the motor, get the efficiency at the horse power thus found. Multiply the horse power by the efficiency and the product will be the power supplied to the pump.

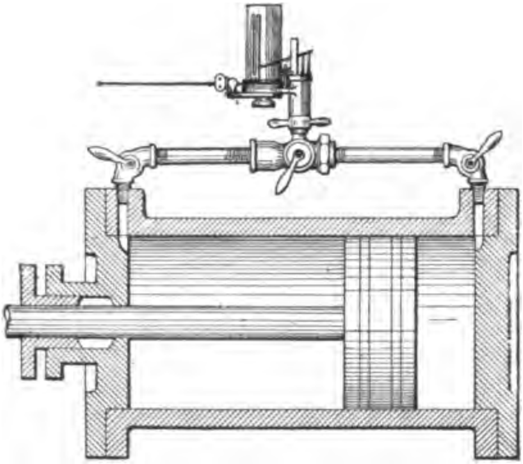


Fig. 72.

The power delivered by it in the form of water pumped is found by the same method as for the belted type.

For a steam-driven pump it is necessary to attach an indicator to the steam end in order to measure the power supplied by the steam. The indicator is connected as shown in Fig. 72, to register the pressure in the cylinder at each point of the stroke. The average of these pressures is then found, measuring between the lines indicating the steam and exhaust pressure as shown in Fig. 73. The sum of lines 1, 2, 3, 4, etc., divided by the number of lines gives the average length, and this multiplied by the scale of the spring, or pounds pressure represented by an inch of height, will give the average effective pressure. The scale of the indicator spring is always found stamped on the cap at one end of it. This average pressure multiplied by the area of the piston in square inches, by the length of the stroke in feet and by the number of double strokes made per minute, gives the work done per minute in one end of the cylinder. In the same way pressure times area, times length, times number of strokes, gives the work for the other end, and the sum of these amounts divided by 33,000 gives the horse power developed. It should be noted that there is a difference between the areas of the two ends of the piston owing to the insertion of the piston rod in one end.

Sometimes the work done in the water cylinder is measured by using the indicator in the same way as described for the steam end. The work done by the water piston can, of course, be found by this method; but there is nothing to show whether the work is used in pumping water or in slip and leakage, except that a slow seating of the valves may show in a reduced average pressure from the diagram, as in Fig. 74.

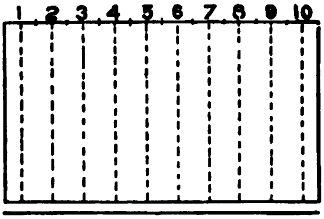


Fig. 73.

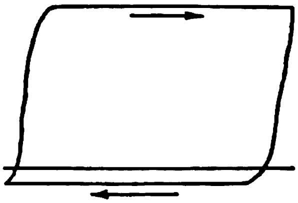


Fig. 74.

To test the leakage past the piston, one cylinder head may be removed and the pump run single acting, the water which passes the piston into the open end of the cylinder being caught and weighed. This does not, however, measure the leakage due to faulty valves or seats, and the only way this can be found is by measuring the water actually pumped.

The slip is then found as follows: Multiply the area of the water piston by its length of stroke and by the number of single strokes if single acting, or double strokes if double acting—all dimensions being taken in feet—to get the number of cubic feet of water which would be pumped per minute as obtained by measurement; subtract the water actually pumped and divide the remainder by the computed volume which should be delivered.

The quotient expressed as a percentage is the slip. Slip is due to the leakage past the piston plunger and through the valves, and the amount varies with the condition of packing and seats and with the promptness of valve closure. It can be kept at a minimum by careful attention to packings, and by running slowly and steadily to allow the valves time to seat.

The efficiency of the pump may be expressed according to various standards. If efficiency as a machine, or mechanical efficiency, is desired, it is found by dividing the horse power utilized in pumping water, by that furnished to the pump by belt, motor or steam cylinder. If efficiency of the water end is wanted, it is found by subtracting the percentage of slip from 100. The total efficiency is the mechanical efficiency multiplied by the pump efficiency.

To find the "duty" the work done in a given time is found by one of the processes already indicated; the heat furnished during that time is computed from the coal burned and the efficiency of the boiler, or better, by condensing the steam used in a surface condenser, weighing it and calculating the heat needed to evaporate that amount of steam, starting with water at the temperature of the exhaust steam. Reduced to the form of equations this becomes:

$$\text{Duty} = \frac{\text{Foot pounds of work done.}}{\dfrac{\text{Heat units used}}{1{,}000{,}000}}$$

Foot pounds work = weight of water multiplied by equivalent head overcome. Heat units used = Pounds of steam used × (total heat at initial pressure minus the heat of the liquid at exhaust pressure).

The values inside the parenthesis must be obtained from tables of the properties of saturated steam, which are given in "Boiler Accessories," works on the steam engine, or engineering handbooks.

For a power-driven pump the duty is found by the formula:

$$\text{Duty} = \frac{\text{Foot pounds of work done.}}{\dfrac{\text{Foot pounds furnished the pump.}}{778 \times 1{,}000{,}000}}$$

since 778 foot pounds are equivalent to one heat unit.

For a motor-driven pump, the foot pounds are equal to average volts × average amperes × time in hours × 2,654.2.

PUMPING BY COMPRESSED AIR.

A system of pumping water by compressed air was known early in the nineteenth century, but no practical working apparatus was devised until Dr. J. G. Pohlé took up the matter in the early seventies.

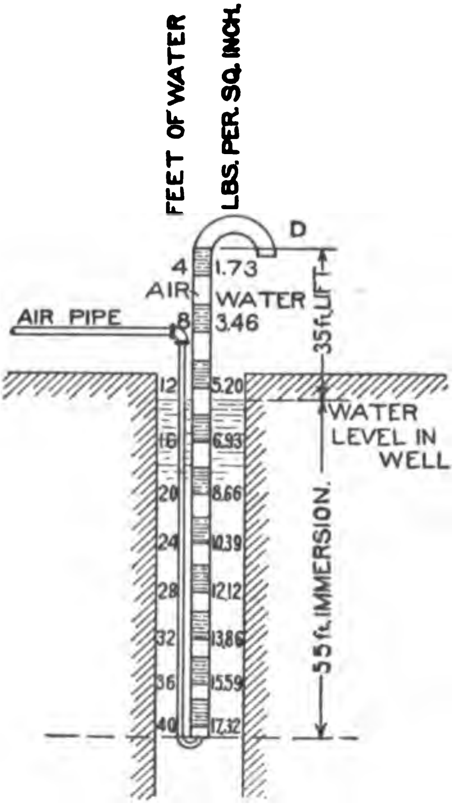


Fig. 75.

The principle of operation is shown in Fig. 75. A discharge pipe, D, is let down into the well so that its lower end is deep below the surface. Compressed air is forced down through a smaller air pipe, and is liberated inside the lower end of the discharge pipe at a high pressure. The air thus set free forms a big bubble the full size of the interior of the discharge pipe, displacing an equal volume of water, and thus making the weight of the column of water and air inside the pipe less than an equal volume of water outside the pipe. The column inside the pipe will therefore rise; and since the formation of air bubbles is continuous so long as the compressed air is supplied, a stream of slugs of water and bubbles of air will rise through the discharge pipe and flow out at the upper end.

The air pressure used must at first be sufficient to overcome the head of water in the discharge pipe, as well as the friction against the sides of the pipe. Also the stream of water must leave the discharge pipe with some considerable velocity.

The pressure needed bears a ratio to the pressure due to the total head of lift and immersion varying from 0.77 for comparatively shallow wells (100 feet or so deep) yielding large volumes of water, to 0.62 for wells 500 feet in depth and having but a small flow. Obviously the lowest air pressure which will do the work is the most economical, since any excess of pressure is used up in producing unnecessary velocity of discharge at the outlet.

The efficiency increases as the ratio of submergence of the lower end of the discharge pipe below the water-level to lift above the water-level increases. There is much disagreement as to the efficiencies that can be obtained; but tests seem to indicate that an efficiency ranging from 18 per cent (for a ratio of 1.5 submergence to lift) up to 37 per cent (for a ratio of 2.6), the efficiency being based on the power required to compress the air, is as much as can be expected.

A great deal depends on the proper proportioning in size of air pipe to discharge pipe, and of submergence to lift. A cross sectional area of water pipe 6¼ times that of the air pipe has been found satisfactory; and a submergence of twice the lift is common in practice, though some tests seem to indicate that a ratio of submergence to lift of 3 to 1 would give better results. The ratio should not in any case be less than 1.5 to 1.

In this connection it should be remembered that the water-level when pumping will always be lower than when at rest, and the above ratio should be fixed for the level under working conditions.

If the air pipe be too large, more air will be furnished than is needed and the discharge velocity will be too great; if the air pipe be too small the air bubbles will not expand sufficiently to fill the discharge pipe but will rise through the water without pushing it along upward.

A velocity not exceeding 20 feet per second is recommended in the air pipe.

The machinery necessary for an air-lift system comprises the following: an air compressor of size sufficient to give the needed air supply, a reservoir large enough to break up the pulsations from the compressor and steady the flow, gauges and valves for regulating the pressure, and the necessary well piping.

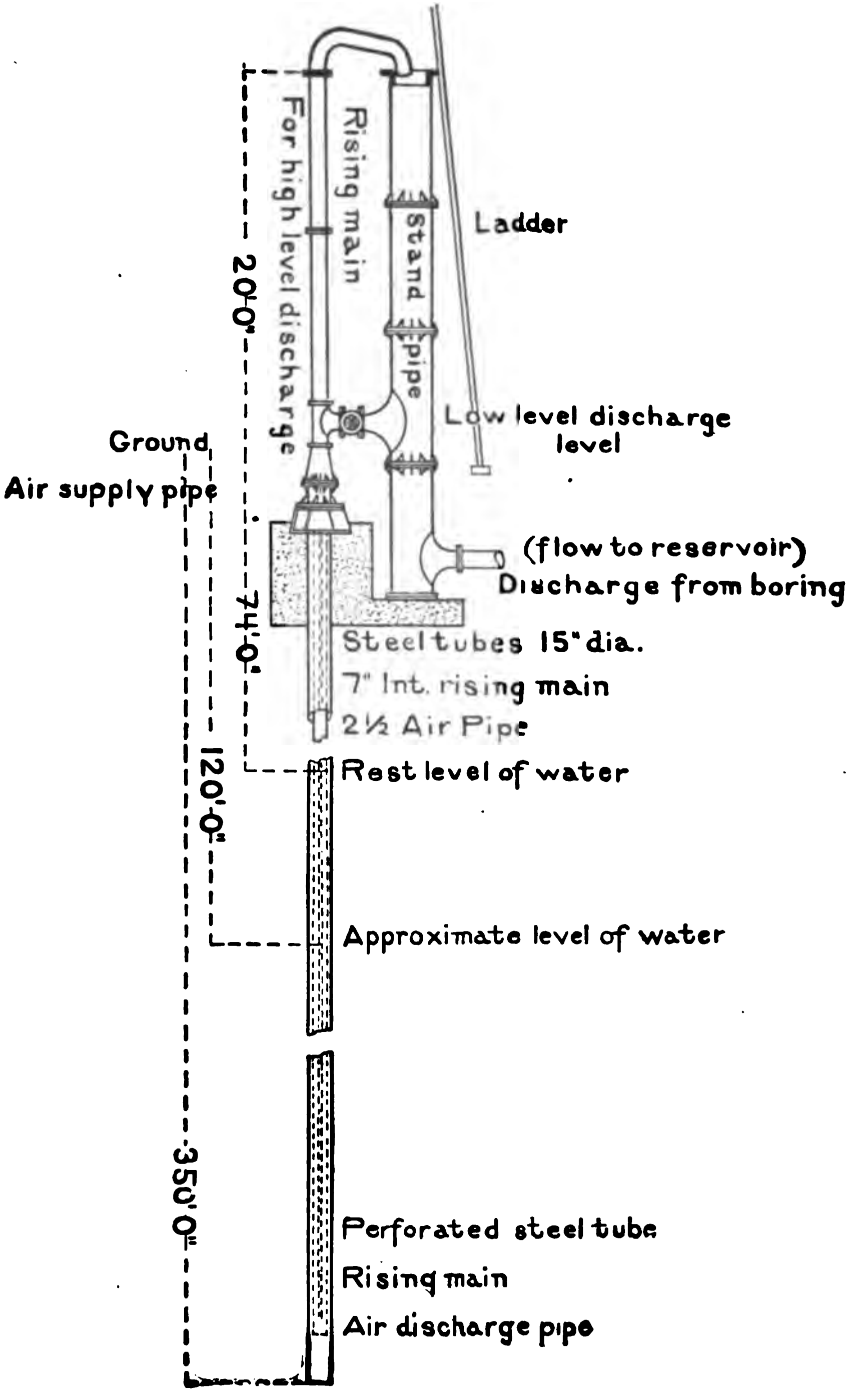

For high level discharge
Rising main
Stand pipe
Ladder
20'0"
Low level discharge level
Ground
Air supply pipe
(flow to reservoir)
Discharge from boring
74'0"
Steel tubes 15" dia.
7" Int. rising main
2½ Air Pipe
Rest level of water
120'0"
Approximate level of water
350'0"
Perforated steel tube
Rising main
Air discharge pipe

Fig. 76.

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For pressures up to 60 pounds, a single-stage duplex compressor will answer; for pressures between 60 and 300 pounds a two-stage compressor, with an inter-cooler between cylinders, will be found economical; while for pressures above 300 pounds a three-stage compressor should be used.

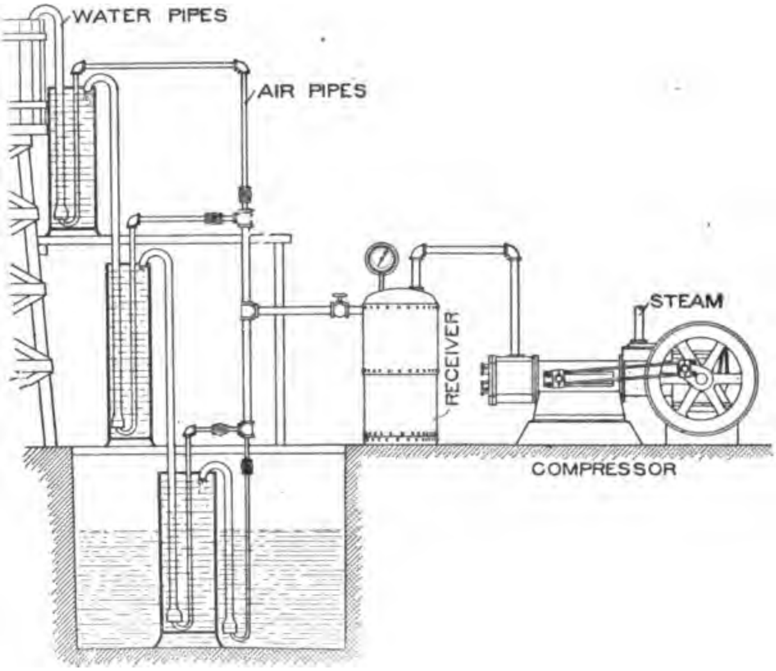


Fig. 77.

The supply of air needed will vary with the lift, and of course with the quantity of water handled. Wm. H. Maxwell gives the following equation:

$$V = \frac{G \times L}{20}$$

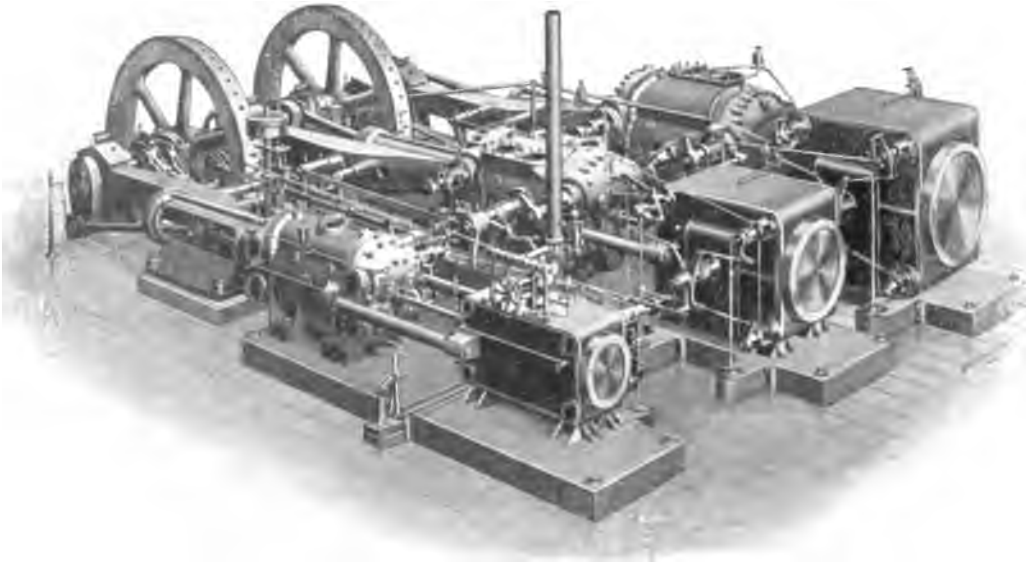
where V = cubic feet of free air per minute.
G = cubic feet of water per minute.
L = lift of water in feet.

The arrangement of air pipes and discharge pipes is largely one of convenience. The least friction will be produced by admitting the air to the bottom of the discharge pipe as shown in Fig.

75; but in deep wells this is often inconvenient, and the air pipe is carried down inside the water pipe as in Fig 76.

For high lifts it is sometimes convenient to use multiple-stage arrangements as shown in Fig. 77, the object being to secure sufficient depth of immersion without the expense of drilling a very deep hole. Only one air compressor is needed, as the pressure can be controlled by throttling the valves for the lower stages without serious loss of economy.

The advantages of the air-lift system are its simplicity, its concentration of all machinery into one place which can be conveniently located, and its ability to handle water containing grit, stones or ashes without injury to the machinery. On the other hand the efficiency is low, and a great depth of well in proportion to the lift is needed. The air-lift system is not adapted to all classes of service, but for handling a number of scattered wells, or a single deep well in an awkward location, it is simple and effective.



GENERAL VIEW OF THREE-STAGE TRIPLE EXPANSION AIR COMPRESSOR.
Nordberg Manufacturing Co.

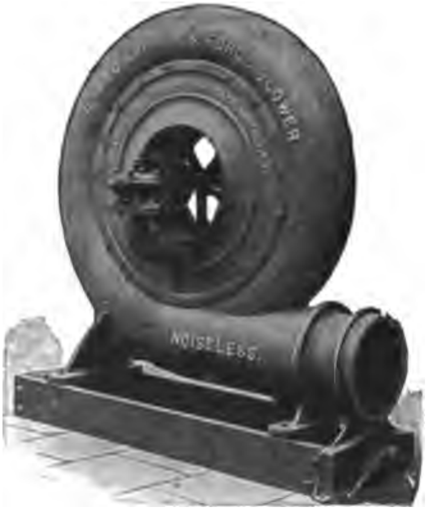
AIR COMPRESSORS

The various practical applications of compressed air cover a range of pressure from a fraction of an ounce to several thousand pounds per square inch. To produce the required supply of air with proper pressure and economy, numerous types of compressors, each having its peculiar and appropriate field, have been designed. These are, in the order of the pressure produced:—the disc fan, the centrifugal blower, the positive blower, the blowing engine, the direct-acting compressor, and the compressor with crank and fly-wheel. By the operation of all these, the same result is produced—the delivery of a quantity of air under an increased pressure and correspondingly increased temperature, and the subsequent rapid loss by the air of its excess of temperature, with a proportionate decrease in volume. Most of the problems in the design of compressors are created by the heating of the air during compression and by the effects so produced upon the apparatus.

Disc Fan.

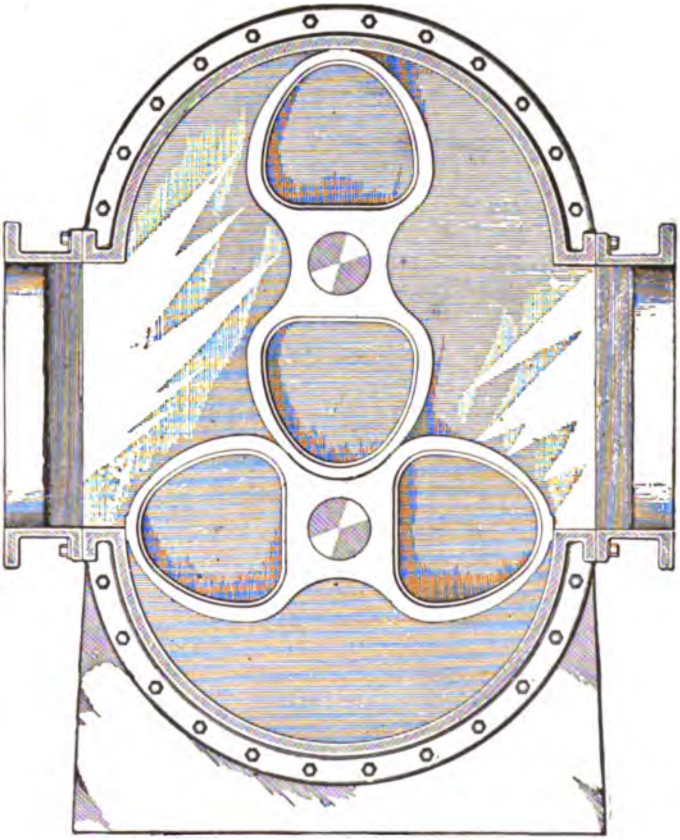
The Disc Fan, corresponding to the marine screw propeller in action, is useful for producing a flow of air of considerable volume but with almost inappreciable increase of pressure. It is thus suitable for purposes of ventilation in expelling foul air, gases, or smoke, or for removing dust, fine shavings, and waste particles from woodworking and grinding machines. The power expended in revolving the fan goes in great part towards whirling the air current around in the fan casing and pipe—a waste of effort which limits the application of this device to conditions of moderate speed and very light pressure.

The Centrifugal Blower. By taking advantage of the tendency of revolving particles to fly away at a tangent, a fan blower of better efficiency and a wider field of usefulness is afforded. Particles of air whirled around by the blades of the disc fan merely fly out and press against the casing; but the centrifugal blower converts this action into useful compression through the provision of a spiral casing around the fan, in which the air whirled off by the blades of the fan is deflected and conducted into a discharge pipe leading off from the casing. The blower therefore consists of a fan with radial or sometimes curved blades which stand parallel to the axis instead of obliquely as in the case of the disc fan, and which revolve in a circular or spiral casing surrounding the fan and opening into a large discharge pipe that leads off at a tangent in any desired direction—up, down, or sideways. Air enters around the shaft of the fan through openings in the center of the casing, and, being received by the blades, is whirled around and thrown outward into the surrounding chamber with a pressure proportionate to the square of the speed of the fan. The pressure thus produced is sufficient for blowing fires of all kinds, for removing small fragments of wood as well as finer particles from wood-cutting machines, for delivering illuminating gas to holders, and for other purposes requiring a pressure of not much over one pound per square inch. The centrifugal blower is the most efficient contrivance known for producing these pressures, but its economy falls off very fast if the speed is made so great as to produce a pressure per square inch of much over one pound.



Centrifugal Blower.

This machine seems peculiar in its behavior owing to the fact that it requires less power to maintain a certain speed when the discharge passage is partly closed than when it is fully open. It is evident that when there is no discharge opening, the fan, together with the air between its blades, will spin around freely like any wheel. If the discharge gate be opened, air will flow about the fan blades to their outer ends, and so into the casing and thence to the pipe system; while air from the center of the fan follows out into the space between the blades, its speed increasing as it approaches the ends of the blades, whereupon it is hurled into the casing. It is the work done in accelerating the motion of these particles of air, which explains the resistance to the movement of the fan; and this work is evidently in proportion to the quantity of air that passes along the fan blades, being zero when, by reason of the closing of either suction or discharge opening, no air is discharged, and being greatest when, with full opening, the volume of air passing is the maximum. The blades of the blower would fly in pieces before the speed could be increased sufficiently to produce an air pressure much over one pound per square inch.



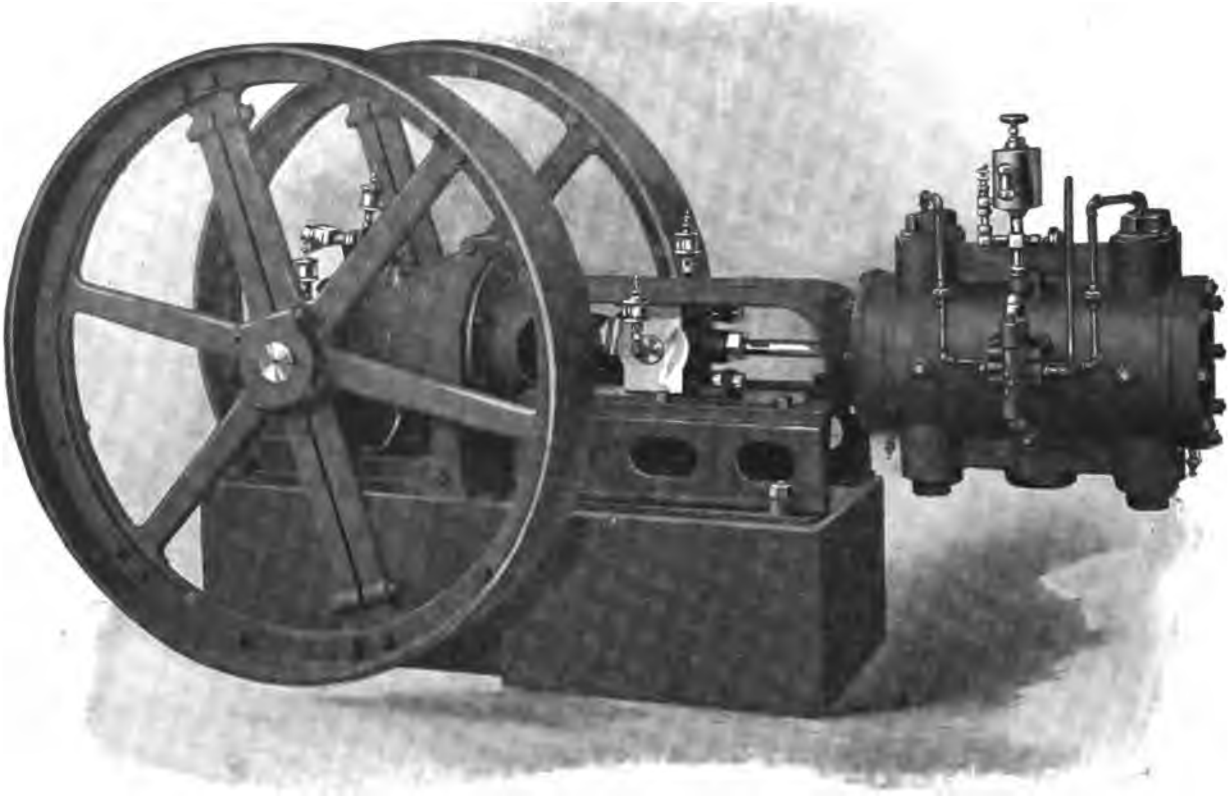
Cross-Section of Positive Blower.

An apparatus identical in principle is the centrifugal pump, which, handling far heavier fluids, produces by centrifugal action pressures measured by pounds instead of ounces.

The Positive Blower. In direct competition with the centrifugal fan is the positive blower. Consider a fan having opposite blades, revolving in a closed casing, and carrying around with it the air confined between the blades. If a radial partition could be slipped in just after one of the blades had passed, the following blade would compress the air between it and the partition, and would force it out through a discharge pipe if one were provided near the supposed partition. This partition might be imagined to be momentarily withdrawn as each blade passed, so that successive portions of air would be trapped and compressed. The positive blower consists of a fan of this sort, with one or more blades, revolving in connection with an "abutment" performing the office of the partition just alluded to. A well-known form consists of a cylinder carrying two radial blades, and revolving in contact with a cylinder of half the diameter which is provided with a gap of proper form to receive the passing blades. As each blade travels around the casing, it drives the air before it up to the smaller cylinder, whose gap comes around at the proper instant to allow the blade to pass, while intercepting the air continuously.

The Air Compressor is a cylinder-and-piston machine like the common steam engine. It comprises two sets of valves, usually designed to be opened automatically by excess of pressure under them and to be closed by gravity or by the action of springs when the pressures become equal. The inlet valves open just after the piston commences its stroke, when the expansion of the compressed air remaining in the cylinder behind the piston has lowered the pressure above the valves. They close at the end of the intake stroke, just as the piston comes to rest. The outlet valves lift during the compression stroke, at about the time the rising pressure in the cylinder becomes equal to that in the outlet passage above the valves; and they close when the flow of air ceases as the piston completes its stroke.

Perfectly correct action of automatic valves is not realized in practice. The valve must evidently be larger than the opening in its seat, so that the upper surface is larger than the area underneath reached by the lower pressure; consequently the valve will not open until the pressure below is greater than that above. To prevent de-



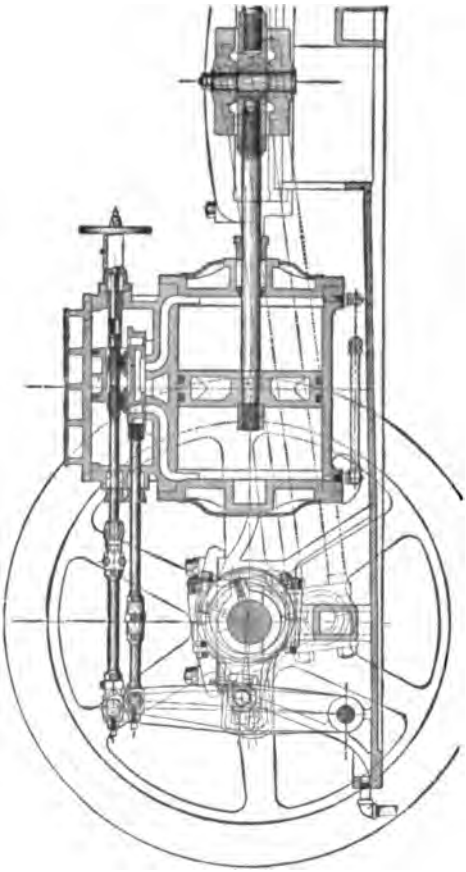
BELT-DRIVEN AIR COMPRESSOR.
Sectionalized for Mule-back Transportation.
Sullivan Machinery Company

structive slamming of the valves, springs must be provided to force them to their seats just as the flow ceases and before the reverse stroke of the piston can cause much backward flow of air. The pressure of the springs acts to choke the flow through the valves, increasing the resistance they offer to the passage of air. Large compressors are therefore often provided with mechanically-actuated valves which are opened and closed smoothly at the proper moment by eccentrics and valve rods. Any of the steam-engine valve gears may be used for compressors, and designed by the same methods, observing only that the compressor is in every way a reversed steam engine, so that its discharge port and valves are duplicates of the inlet details of the engine, while the engine exhaust and the compressor inlet valves are also similarly related.

Varying initial (or boiler) pressure is compensated, except in throttling engines, by varying time of inlet-valve closure or "cut-off;" and varying discharge pressure in a compressor calls for variation in time of opening of the discharge valves. In both cases, the means of variation constitute the chief problem for the designer.

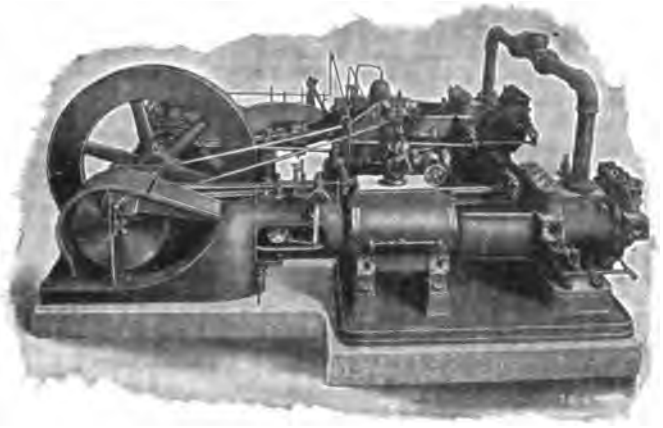
Air Compressor.

Mechanically-moved inlet valves of compressors act always at the same points, opening a trifle after the piston starts on the intake stroke, and closing exactly at the end of the same stroke; but the discharge valves must open at the instant the piston has compressed the air in front of it to a pressure equaling that above the valves in the discharge pipe, and must close always at the same instant, at the end of the stroke. As the compressor may be working against a pressure greater or less than that regularly carried, the discharge valves must be so controlled as to open at whatever point is required by the pressure then being carried. The requirements are sometimes met by putting automatic lift-valves above, or even directly upon, the mechanical discharge valves, giving the combination the effect of valves automatic as regards their opening, but positively closed by the mechanism at the proper instant. Such lift-valves, being shut off entirely at the proper closing instant, seat themselves without noise or shock, and may therefore have very light springs, causing less resistance to the air passing through.


Sectional View of Steam Cylinder of Air Compressor.

Where the expense of full mechanical action is warranted on account of the superior efficiency obtainable, poppet or rotary valves may be arranged to open by means of springs or air dashpots. The opening device is released either through the rising pressure in the cylinder easing the valve on its seat, and reducing the friction until the valve, when balanced, slips freely open; or through the same pressure acting on a piston attached to a pusher, the operation of which results either in starting the valve in spite of friction or in lifting a catch and thereby freeing the spring or dashpot mechanism of the compressor.

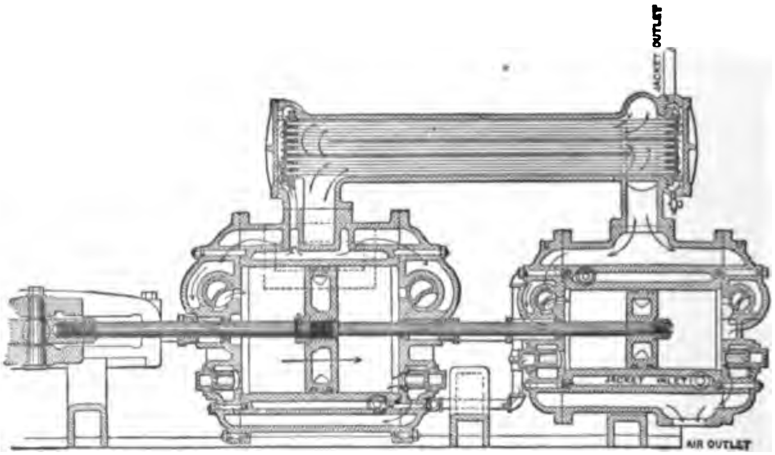
Compound Compressors. While low-compression pressures are accompanied by only moderate heating of the air during compression, the production of high pressures is attended with excessive heat and considerable increase in the volume of the compressed air. As the air leaving the cylinder soon resumes the normal temperature, and decreases in volume accordingly, the extra work done in compressing the increased volume is wasted. Compressing cylinders in operation are always cooled by water or otherwise; but it is impossible, even by spraying water into the cylinder, to keep the air from rising considerably in temperature. For high pressures, resort is therefore had to compound compression, the air being passed successively through larger low-pressure to smaller high-pressure cylinders, between which are located inter-coolers whose function is to restore the air to its original temperature before it enters on the next stage. The volume of the air is thus kept as small as possible; and the successive stages of compression result in producing the required pressure, with a minimum of loss from heating during the process. Two-stage machines are preferred to single-stage where air must be compressed to one-sixth or a greater fraction of its volume at atmospheric pressure (measuring pressures from absolute vacuum); and three or more stages are required in compressing to less than one-sixth. Cylinder diameters are



Cross Compound Duplex Air Compressor.

selected which will provide for about the same amount of work being done in each cylinder.

Most compressor problems deal with air taken directly from the atmosphere at its sea-level pressure; but, as at moderate elevation there is a marked decrease of the atmospheric pressure, compressors for high locations must deal with air at pressures below fifteen pounds absolute. Under such conditions the volume of air taken into the compressor at each stroke weighs less, and therefore less air is delivered by the compressor, while there is a corresponding decrease in the power to run the machine. The ratio of compression, and the rise in temperature, are proportionately increased, so that a two-stage compressor may be desirable for pressures that would call for only single cylinders at sea-level. Thus, at many of the mines in the Rocky Mountain region, the atmospheric pressure is as low as eleven pounds per square inch, so that 90 pounds' air pressure by gauge requires a compression ratio of nine to one, which is considerably beyond that proper for a single-stage compressor. In general, high-level compressors should be specially proportioned for their work.



Sectional View of Air Cylinder and Inter-Cooler.

Methods of Driving. Like pumps and other machinery, compressors are direct-connected to engines or are driven through gears or belts from separate sources of power. The reciprocating piston compressor requires a varying effort to balance the cylinder pressure, since, during the stroke, the piston moves against an increasing air pressure, and finally against the full discharge pressure, in pushing out the contents of the cylinder. Direct-connected compressors are either "straight-line" (tandem), having steam and air pistons on the same piston rod, or they are connected to cranks set at an angle on a common shaft. The first method reduces floor space and cost, but requires very heavy fly wheels, and makes the machine liable to stop on a center if run much below full speed and capacity.

It is evident that steam used expansively supplies against the piston a force decreasing toward the end of the stroke, while air during compression opposes a force increasing towards the end of the stroke; thus the power rapidly falls off as the resistance increases, causing a perceptible reduction in speed at the end of each stroke. If such a machine could be run at high speed, the weight (or more correctly, the mass) of the pistons and connections would, by inertia, help out the decreasing steam pressure when slowing to pass the centers, and thus produce a more even effort on the crank; but sufficiently high speeds are not possible for the automatic-lift valves

Portable Motor-Driven Compressor.

generally used on small compressors. The varying power and resistance can be very satisfactorily balanced by connecting steam and compressing pistons to separate cranks set at right angles. Having provided two frames and cranks, a slight additional outlay will supply an extra pair of cylinders tandem to the first pair, making a full duplex compressor. The excess steam pressure at the commencement of the stroke of one side is here transmitted by the crank shaft to the other side of the machine, to help out the deficient pressure of the expanded steam when the stroke is nearly finished. Such a machine has no "dead centers," and can be run at very slow speed when necessary.

As it is generally desirable to maintain a constant air pressure, and to vary the speed of the machine according to the quantity of air required, speed governors for the steam cylinders are not needed except to prevent racing in case of a bursting pipe or other excessive discharge of air. Some form of adjustable cut-off valves is very desirable in order to allow of suiting the work of the steam cylinder to the load. The pressure is controlled by automatic devices actuated by the rise and fall of the air pressure, either shutting off the air intake, opening a by-pass around the compressor piston, or (in case of duplex machines which can start from rest without attention) shutting off steam and stopping the machine.

A description of the standard types of commercial compressors would be incomplete without reference to the most remarkably wasteful "steam eater" known to the compressor trade—a machine using ten times as much steam as would be necessary for pumping the same amount of air by means of a fairly economical compressor, and yet a device most ingenious and entirely satisfactory for its work. This is the air-brake pump, which, for actual conditions of train service—where it stands idle until the closing of the throttle and the application of brakes leave a large and heavily fired steam boiler to blow off at the safety valve until the fire can be checked—is seen to be well adapted. Indicator cards show that the entering steam is throttled through about half the stroke, while the exhaust is similarly choked at first and only let out freely about the time of full opening of the valve. The result is a "straight-line" compressor having no crank or fly wheel, with nothing moving but its two pistons and one rod, and yet so perfectly balanced between effort and resistance that its strokes are smoothly made at any speed from slowest to fastest, and all with max-

.mum simplicity and minimum weight. These machines are also built with compound steam and two-stage air cylinders, and in these cases have pressures in the cylinders so nearly uniform that the steam distribution may be considerably more economical than it is possible to obtain in the single-stage compressor.

The selection of general types and special details for compressors must be based, as in all other cases of machine design, on the practical condition of getting the best return from the money invested, with due regard to the importance of reliability and durability in each particular case.

REVIEW QUESTIONS.

PRACTICAL TEST QUESTIONS.

In the foregoing sections of this Cyclopedia numerous illustrative examples are worked out in detail in order to show the application of the various methods and principles. Accompanying these are examples for practice which will aid the reader in fixing the principles in mind.

In the following pages are given a large number of test questions and problems which afford a valuable means of testing the reader's knowledge of the subjects treated. They will be found excellent practice for those preparing for College, Civil Service, or Engineer's License. In some cases numerical answers are given as a further aid in this work.

REVIEW QUESTIONS

ON THE SUBJECT OF

CONSTRUCTION OF BOILERS.

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1. If they are near together how are two flat parallel surfaces stayed?

2. Describe a rivet.

3. Since copper is such a desirable metal for boiler work why is it not used more extensively?

4. Why is a large factor of safety used for stays?

5. State what you can (briefly) about the injuries done to plates by punching and the methods employed to overcome them.

6. Why are not welded joints used more generally?

7. In what two ways are tubes fastened to the tube sheet?

8. About what is the ratio of length to diameter of the multitubular type of boiler?

9. Explain riveting with countersunk head.

10. Is the greatest tendency to rupture along the longitudinal or the circumferential seams?

11. Why is the length of a grate limited?

12. Which is the stronger form of riveting, the lap joint or double butt joint (both with double riveting)?

13. Why are the short screw stay bolts turned smooth in the center?

14. Why are flanged joints preferable to those made with cast iron angle irons?

15. What is the water-leg?

16. For what qualities are boiler materials tested?

17. What is the principal advantage of pneumatic calking?

18. Name some of the considerations that must be kept in mind when designing a boiler?

19. If the rivet is 1 inch in diameter what would *you* make the pitch, if single riveted lap joint?

20. Upon what does the choice of type of boiler depend?

21. What is the thickness of the shell of a boiler if the diameter is 60 inches, the steam pressure 70 pounds per square inch, the joint has an efficiency of .68 and the working strength of the metal 9000 pounds per square inch? The working strength equals the breaking strength S, divided by f.

Use the formula $t = \frac{fDp}{2SE}$. Ans. $\frac{3}{8}$ inches.

22. Find the allowable pressure, if the above conditions are the same, with the exception of t, which is $\frac{1}{2}$ inch.

Ans. 102 pounds per square inch.

23. Why should calking be done carefully and with a proper shaped tool?

24. About what fraction of the volume of a multitubular boiler is the steam space?

25. If the boiler is to supply steam to a high speed engine, should the steam space be larger or smaller than if to a low speed engine?

26. Why is wrought iron preferable to cast iron for boiler shells?

27. Why is a riveted joint weaker than an uncut plate?

28. A boiler evaporates 3211 pounds of water per hour. The grate is $6\frac{1}{2}$ feet by 4 feet and the coal evaporates $9\frac{1}{2}$ pounds of water per pound. What is the rate of combustion?

Ans. 13 pounds per sq. ft. per hour.

29. What kind of joint is generally used for longitudinal seams?

30. What should be the diameter at the top of a chimney 81 feet high if 2.88 pounds of coal are burned per horse-power per hour, and the horse-power is 160?

Use formula foot page 56. Ans. 28 inches.

31. Name four general requirements that a boiler must have.

32. What are the advantages of machine riveting

REVIEW QUESTIONS

ON THE SUBJECT OF

TYPES OF BOILERS.

1. Why are water-tube boilers lighter than those of the fire-tube type?

2. In the fire-tube boiler, are the tubes large or small for forced draft? Why?

3. Make a sketch of a multitubular boiler and locate the important parts. Show by means of arrows the path of the hot gases.

NOTE: The sketch should combine Figs. 12 and 13.

4. Name three boilers that have curved water tubes, are non-sectional, and have a steam drum of the cross type.

5. Describe briefly the Stirling boiler.

6. Classify, under the headings "sectional" and "non-sectional," all the water-tube boilers described in this Instruction Paper.

7. Trace the changes that occurred in the development of the horizontal multitubular boiler from the plain cylindrical.

8. What is the peculiarity of Beggs' Directurn Boiler?

9. What are the advantages of the double-ended return-tube boiler?

10. Trace the path of the hot gases in the Atlas boiler. For what purpose is the third drum?

11. In what way does the Niclausse boiler differ from all others here described? Trace the course of the water. Describe the construction.

12. Describe briefly the flue boilers,—Cornish, Lancashire, and Galloway. Identify each by stating the peculiarities in a few words.

13. What is a Breeches-flued boiler?

14. Sketch the longitudinal section of a single-furnace single-ended return-tube boiler and name the principal parts.

15. Is the Babcock and Wilcox a sectional boiler? In what way does it differ from the Root?

16. What is the chief peculiarity of the Harrison boiler?

17. How does the Belpaire fire-box differ from the Wootten?

18. Why are vertical fire-tube boilers not economical? Describe the fire-box construction in the Manning boiler?

19. Describe the feed water heater used with the Mosher boiler. Is this a sectional boiler?

20. What two boilers belong to the class that may be briefly described as:

Straight Vertical Water-Tubes, Single-Tube, Non-Sectional.

21. Describe the Robb–Mumford boiler. Why is the furnace flue corrugated? Why is the lower shell inclined?

22. In what ways does the Hazelton boiler differ from the Wickes?

23. Why should the boiler shown in Fig. 34 be economical and at the same time occupy but little space?

24. Name all the boilers you can think of that have the following characteristics:

Water-Tube—Non-Sectional—Straight-Tube—Water-Tubes Horizontal Or Nearly So—Steam Drum Horizontal.

25. Is the Worthington boiler sectional? What can you say regarding the extent of the furnace as compared with the tubes?

26. Discuss briefly some of the advantages claimed for water-tube boilers.

27. Why should one of the brackets supporting a multi-tubular boiler be placed on rollers?

28. In general, how are flat plates within a boiler prevented from bulging or collapsing? How are two flat or curved surfaces stayed when they are close together?

29. The Cahall boiler may be described as follows:

Water Tubes Vertical—Straight-tube— Non-Sectional

Why cannot the Milne be placed in the same class?

30. In what boiler are small circulating tubes used for obtaining dry steam?

REVIEW QUESTIONS

ON THE SUBJECT OF

BOILER ACCESSORIES.

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1. When are check valves used? Explain the action.

2. What is the unit of boiler horse-power?

3. What devices are used to drain condensation from steam pipes?

4. What are the defects of the lever safety valve?

5. Why should the feed supply be regulated so that the water level shall be as nearly stationary as possible?

6. Describe the principle upon which the injector or inspirator works.

7. Why is it necessary to reduce the actual evaporation to an equivalent evaporation, before comparisons can be made?

8. Name some methods for preventing smoke.

9. Describe the two types of feed water heaters.

10. Of what are fusible plugs made? Where are they placed?

11. What are the requisites of a good boiler setting?

12. What portions of a boiler should be kept clean?

13. What is a good average heat value for coal?

14. In building a fire 360 pounds of wood were used. What is the coal equivalent?

15. State some reasons why the efficiency of a boiler and furnace is considerably less than 100 per cent.

16. Describe the pop safety valve.

17. What should be the diameter of a safety valve when

the boiler evaporates 1.346 pounds of water per second, if the lift is .1 of an inch and the pressure 100 pounds (absolute)?

Ans. 3 inches.

18. Describe the process of banking a fire.

19. What harm is caused by the admission of engine oil to the boiler?

20. In general, how are boiler explosions prevented?

21. What are the three kinds of firing?

22. Name and state the objects of the two kinds of boiler tests.

23. Describe the action of the down draft furnace.

24. Where is the feed water generally introduced?

25. Name some good pipe coverings.

26. Why is scale undesirable?

27. Why should the steam space never be exposed to the heat of the fire?

28. A lever safety valve is set to blow off at 65 pounds. The ball at the end weighs 110 pounds, the lever weighs 48 pounds and has its center of gravity 18 inches from the fulcrum. The valve is $4\frac{1}{2}$ inches from the fulcrum. The valve has a diameter of $4\frac{1}{2}$ inches and with the spindle weighs 14 pounds. At what distance from the fulcrum must the ball be placed?

Ans. $33\frac{7}{8}$ inches (nearly).

29. Describe in detail the methods for starting a fire in a boiler furnace, for bituminous coal.

30. Why are patches of scale more harmful than a uniform coating?

31. If you were in charge of a boiler and the gage cocks gave no indication of water, what would you do?

32. Describe briefly the two most important kinds of coal.

33. Show why it is economical to introduce the feed water at a high temperature. What per cent. is gained if the feed water enters at 200° F. instead of 60° F.? The pressure is 110 pounds (gage).

34. Describe the action and principle of the Bourdon gage. (Make sketch).

35. Before opening the drafts, what should a fireman do?

REVIEW QUESTIONS

ON THE SUBJECT OF

STEAM PUMPS.

1. Find the pressure per square inch corresponding to a head of 187.42 feet.

2. Under what conditions should a centrifugal pump be used?

3. Describe the rotary pump.

4. If a large quantity of water is to be lifted a slight amount (for instance 20 feet) is it better to use a direct-acting pump or a centrifugal pump?

5. How many gallons per minute can be raised by a pump having a water cylinder 8 inches in diameter, 12 inches stroke, if it makes 45 double strokes per minute? Assume 6% slip.

6. What materials are used for valve discs for cold water? For hot water?

7. Which is the better form of spring for a valve, cylindrical or conical? Why?

8. Under what conditions is it better to use a plunger pump rather than the piston type?

9. A single cylinder pump is 6"x10". What should be the volume, diameter and height of the air chamber on the discharge pipe?

10. A 12"x10" duplex makes 52 double strokes per minute. What is the piston speed in feet per minute and what is the discharge in gallons?

11. A double-acting pump is to discharge 232 gallons per minute. On account of the pressure against which the pump must work, the water cylinder must be 7 inches in diameter. The allowable number of double strokes per minute is 55. What is the length of stroke? What is the piston speed?

12. Suppose a pump has a suction of 14 feet and the loss due to friction in the piping is 3 pounds per square inch. The pump is a 3" × 10" and the discharge is against a head of 86 feet. If the pump makes 40 double strokes per minute, what horse power is required, allowing 20% for friction?

13. The head against which a pump works is 122 feet. If the plunger is 4 inches in diameter and the steam end 7 inches, what steam pressure must be used? Consider the plunger friction to be 75 pounds.

14. A boiler-feed pump works against a pressure of 145 pounds. It draws water from a tank located so that the lift is 16 feet. If the steam pressure at the pump is 130 pounds and steam cylinder is 8 inches in diameter, what is the diameter of the plunger? (a) Make no allowance for friction. (b) Assume friction to be 15%.

15. Describe the setting of the valves of a duplex pump.

16. Describe the valve motion of a duplex pump.

17. Draw a diagram and locate the high-pressure cylinder, the low-pressure cylinder, the pistons, the piston rod, water cylinder, plunger, and air chamber of a tandem compound pump.

18. What means are used for connecting the shaft of a gas or steam engine to a triplex pump?

19. How is steam cushion obtained in a pump?

20. Why are laps added to steam valves?

21. What should be the maximum lift for a disc valve?

22. Explain with sketch the action of the combined forcing and lifting pump (double acting).

23. Describe the usual lost-motion device for a duplex pump. Why is it used?

24. Where should the check valve of a suction end be placed?

25. What kind of packing is commonly used in the water-end stuffing box?

26. If a pump slams on one stroke but not on the other, what is the probable cause?

27. Give method of procedure in starting a compound condensing pump.

28. What is the most common cause of short stroke on one side of a duplex pump?

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